

A PASSIVE COOLING DESIGN FOR MULTIFAMILY RESEDENCES
IN HOT, HUMID CLIMATES

by

Joseph C. Tang
B.S. Tunghai University, 1977
Taichung, Taiwan

Submitted in Partial Fulfillment
of the Requirement for the
Degree of
Master of Science In Architecture Studies
at the
Massachusetts Institute of Technology
June, 1983

c Joseph C. Tang, 1983

The author hereby grants to M.I.T. permission to reproduce
and to distribute publicly copies of this thesis document in
whole or in part.

Signature of Author.....
Department of Architecture
May 6, 1983

Certified by....
Timothy Johnson
Principle Research Associate
Thesis Supervisor

Accepted by..
N. John Habraken, Chairman,
Departmental Committee for

MASSACHUSETTS INSTITUTE
OF TECHNOLOGY

MAY 26 1983

LIBRARIES

A PASSIVE COOLING DESIGN FOR MULTIFAMILY RESIDENCES
IN HOT, HUMID CLIMATES

by

Joseph C. Tang

Submitted to the Department of Architecture
on May 6, 1983
in partical fulfillment of the requirements
for the Degree of
Master of Science in
Architecture Studies.

ABSTRACT

People living in hot, humid climates suffer either from extremely uncomfortable weather conditions or from the great cost of air-conditioning systems for maintaining comfort. Most of the available passive cooling techniques which are applicable to other climates have been proven to be ineffective in hot, humid climates.

This thesis examines the available passive cooling techniques and describes their potential and limitations. This thesis also proposes a series of passive cooling design solutions for the multifamily residences found in hot, humid climates like Taipei. In addition, a low-cost cooling system is developed as a substitute for conventional high-cost air-conditioning systems, to be used during periods when passive means are incapable of maintaining human comfort. This newly proposed system employs a seasonal cooling reservoir coupled to a heat pump DHW heater. The continuous heat extraction from the cooling reservoir for DHW needs makes the reservoir an effective cooling resource in the summer.

Thesis supervisor: Timothy Johnson

DEDICATION

This thesis is dedicated with deep love and respect to my Lord Jesus Christ for His support in wisdom, perserverance, faith, and love; to my dear parents and grandmother for having raised and supported me spiritually and physically; to Charlotte Fan for her continual support in love and prayer.

ACKNOWLEDGEMENTS

Timothy Johnson for support, guidance, and
valuable suggestions

MIT provost office for scholarship

Bill Bartovics for editing assistance

Matthew Toth

Yi-May Chen for typing and formatting

Shih-Kwang Liu

Chen-Yih Hsu for graphic assistance

Chih-Shiang Liu

Lee-Hsueh Hung

Wei-Kang Tsai

Chi-Fang Chen

Ming-Cheng Chen

Weather Bureau, R.O.C. for weather data information

TABLE OF CONTENTS

Chapter I Introduction

1.0	Background	11
2.0	Objectives	12
3.0	Scope	13

Chapter II The Available Passive Cooling Techniques

1.0	Load Control	14
1.1	Orientation	14
1.2	Window and Sunshading Design.....	17
1.3	Choice of Materials and Colors	18
1.4	Ventilation Control.....	20
2.0	Convective Cooling	20
2.1	Physical Effect.....	20
2.2	Natural Ventilation	22
2.3	Thermally Induced Ventilation	23
2.4	Wind Induced Ventilation	25
2.5	Night Ventilation	27
2.6	Mechanically Driven Ventilation	28
3.0	Evaporative Cooling	29
4.0	Radiant Roof Cooling	29
5.0	Earth Coupled Cooling	30
6.0	Detached Earth Coupled Cooling	30
7.0	Dehumidification	32

Chapter III Weather Data and Design Tools

1.0	Weather Data	34
1.1	Geographical Position	34
1.2	Essential Weather Data	35
1.2.1	Air Temperature	35
1.2.2	Humidity	36
1.2.3	Air Movement	36
1.2.5	Other Helpful Data	38
1.3	Hourly Weather Data	38
1.4	Monthly Average Hourly Weather Data (M.A.H.W.D.)	39
2.0	Design Tools	41

2.1	Psychrometric Chart Analysis	41
2.2	Bioclimatic Chart Analysis	43
2.3	Monthly Average Hourly Weather Data Chart (M.A.H.W.D. Chart).....	45
2.4	Sun Chart and Shading Mask Protractor	47
2.4.1	Sun Chart	47
2.4.2	Shading Mask Protractor	52
2.4.3	"Shaded" Sun Chart and Shading Devices Design	58
2.5	Wind Roses	61
2.6	Soil Temperature Profile Analysis ...	62
2.7	Cooling Load Calculations	62

Chapter IV Design Issue and Passive Solutions

1.0	Design Issue	69
2.0	Plan Rearrangement	71
3.0	Load Control	76
3.1	Orientation	76
3.2	Window Design	77
3.3	Sunshading Design	77
4.0	Choice of Materials and Colors	77
5.0	Sealed Building	79
6.0	Variation	79
6.1	Ventilation Schedule	79
6.2	Wind Roses and Design Solutions	84
6.2.1	Main Facades Facing North and South	84
6.2.2	Main Facades Facing East and West.....	86
6.2.3	Main Facades Facing North-East and South-West	86
6.2.4	Main Facades Facing North-West and South-East	90

Chapter V Low-Cost Cooling Using Annual Storage

1.0	Introduction.....	93
2.0	Summer Cooling Load	93
3.0	Heat Pumps	95
4.0	Heat Pump, DHW, and Seasonal Cooling Reservoir	98
4.1	Domestic Hot Water Heater	98

4.2	Seasonal Cooling Reservoir and Sensible Cooling	99
4.2.1	Seasonal Cooling Reservoir ..	99
4.2.2	Reservoir Material	100
4.2.3	Reservoir's Volume and Location	101.
4.2.4	The Amount of Insulation	101
4.2.5	Charge and Discharge Periods	104
4.2.6	Sensible Cooling - Radiant Panel System	106
4.3	Heat Pump and Latent Cooling	107
4.3.1	Heat Pump DHW Heater and Dehumidification	110
4.3.2	Run-Around Coil	112
4.4	Heat Pump Operation Schedule	112
4.4.1	Heat Pump Operation Schedule	112
4.4.2	Annual Reservoir Temperature Profile	114
4.4.3	Mechanical Loops	115
5.0	Economical Analysis	118
5.1	Initial Cost	118
5.2	Energy Savings	119
5.2.1	Domestic Hot Water Saving ...	119
5.2.2	Summer Air-Conditioning Saving	119
5.2.3	Total Yearly Saving	120
5.3	Perspective	120
Chapter VI Conclusions and Recommendations		
1.0	Conclusions	122
2.0	Recommendations	124
Appendices		
A.	Cooling Load Calculations	127
B.	Heat Exchange and Temperature Profile of the Cooling Reservoir.....	137
C.	Ground Temperatures at Different Depth...	147
D.	Economical Analysis.....	148
Bibliography.....		159

LIST OF FIGURES

Figure No.

2.1	General Rule-of-Thumb for Building Orientation.....
2.2	Solar Incidence on the South and West Facades
2.3	Profile Angle
2.4	Typical Wind Pressure Distribution on A House.....
2.5	The Effect of Applying Wind-Inducing Devices
2.6	Sorbent Dehumidification
3.1	Psychrometric Chart
3.2	Givoni's Psychrometric Chart
3.3	Givoni's Psychrometric Analysis of Taipei's M.A.H.W.D.....
3.4	Olgyay's Bioclimatic Chart
3.5	Aren's Bioclimatic Chart.....
3.6	Aren's Bioclimatic Analysis of Taipei's M.A.H.W.D.....
3.7	Taipei's M.A.H.W.D. Chart
3.8	Human Comfort Needs on the M.A.H.W.D. Chart as Analyzed by Givoni's Psychrometric Chart.....
3.9	Human Comfort Needs on the M.H.W.A.D. Chart as Analyzed by Aren's Bioclimatic Chart.....
3.10	Solar Altitude and Azumuth
3.11	Sky-Dome Sun Chart
3.12	Cylindrical Sun Chart
3.13	Sky-Dome Shading Masks with Different Observation Points.....
3.14	Sky-Dome Shading Mask Protractor
3.15	Cylindrical Shading Mask Protractor
3.16	Sky-Dome and Cyclindrical Shading Masks for the Same Shading Devices.....
3.17	Corrections to Olgyay's Shading Masks
3.18	"Shaded" Sun Chart for Taipei
3.19	Wind Roses of Taipei, for the Time Period: 1972-81.....

3.20	Ground Temperature Profiles for Taipei
3.21	Building Heat Gains in Summer
4.1	The Prototype of Present Multifamily Residence in Taipei.....
4.2	Privacy Interruptions in Contemporary Building Designs.....
4.3	A New Design Proposal
4.4	A Design Concept for Preserving the Present Floor Area Per Family.....
4.5	An External Wall Section.....
4.6	Weatherstripping at Window and Door Frame.....
4.7	Ventilation Schedule on A M.A.H.W.D. Chart.....
4.8	Sunshading and Wind-inducing Design for North-South Orientated Buildings.....
4.9	Sunshading and Wind-inducing Design for East-West Orientated Buildings.....
4.10	Movable Insulation with Louvers below The Window Sill.....
4.11	A Second Option for Sunshading and Wind-Inducing Design in East-West Orientated Building.....
4.12	Sunshading and Wind-inducing Design for NE-SW Orientated Buildings
4.13	Sunshading and Wind-inducing Design for NW-SE Orientated Buildings.....
5.1	Typical Unit Plan Facing South and Its Summer Load in Taipei.....
5.2	Flat-Plate Thermosiphoning Solar Heater.....
5.3	Cooling Reservoir as A Floating Foundation for Buildings.....
5.4	The Insulation Effect of Soil Mass around The Reservoir.....
5.5	The Insulation Effect of Soil Mass below The Reservoir.....

5.6	Horizontally Extended Insulation Detail at The Building Edges.....
5.7	The Dew-Point Temperature of A Typical Conditioned Space on A Psychrometric Chart.....
5.8	Radiant Panel Condensation Control with A Four-Pipe System.....
5.9	Detail of Ceiling Panels for radiant Cooling.....
5.10	Basic Components of A Conventional Dehumidifier.....
5.11	Water Source Heat Pump.....
5.12	Conventional Heat Pump DHW Heater.....
5.13	Water Source Heat Pump with A Run-Around Coil.....
5.14	Heat Pump DHW Heater with A Run-Around Coil.....
5.15	Annual Cooling Reservoir Temperature Profile.....
5.16	Mechanical Loops for Non-Cooling Season Operation....
5.17	Mechanical Loops for Cooling Season Operation.....
5.18	The Perspective of Cooling Reservoir Application.....

CHAPTER I

INTRODUCTION

1.0 BACKGROUND

As the energy crisis first appeared in 1974, people began to realize that cheap energy is to be no more, and that total reliance on mechanical systems, capable of controlling an environment, is by no means the optimal design solution for architecture in the future. More and more buildings designed to passively utilize natural energy flows have begun to show up around the world. Most of them are being designed to be passively heated. Passive cooling techniques, on the other hand, have been very poorly developed due to the complexity of the energy exchanges involved. They are particularly complicated in hot, humid climates where high temperature and R.H. are always regarded as the most difficult problems to deal with passively.

Unfortunately, most of the developing countries lie in the hot, humid belts. Their technologies always follow those of the more "advanced" nations in the developed world. The "advanced" passive technologies however, are primarily applicable to passive heating rather than to passive cooling. Consequently, people in third world countries always suffer for these unsolved problems. They must tolerate either discomfort due to weather conditions, or the tremendous cost of using air-conditioning systems to obtain

comfortable interior conditions. As a result, the reduction of energy expended for air-conditioning and the maximum use of passive cooling techniques are the essential parts of any effort made by countries which lie in the hot, humid regions, so that overall building energy savings and human comfort can both be obtained simultaneously.

As the comfort in residences is the primary concern of people in these climates, and as the 5-story multifamily apartment is the most popular type in Taipei (which is chosen to illustrate the proposed techniques), this thesis will focus the passive cooling techniques appropriate to multifamily residential buildings in hot, humid climates.

2.0 OBJECTIVES

- 1) Examine a number of existing passive cooling techniques and indicate their limitations and potentials.
- 2) Develop an analytical method to utilize readily accessible weather data for the design of a passively cooled residential building.
- 3) Introduce a passive cooling building design which can maximize the passive cooling potential.
- 4) Introduce a low cost cooling technique using a heat pump DHW heater coupled to a seasonal cooling reservoir during the periods when passive means are incapable of satisfying the cooling loads.
- 5) Use hourly weather data from Taipei to illustrate the

analytical method, climate responsive design, and the low-cost cooling technique for multifamily residential buildings in hot, humid climates.

3.0 SCOPE

The passive cooling design referred to in this thesis is not intended to change the existing popular building configuration but rather to find a feasible solution which fits this popular type of building. This approach is important in terms of convincing manufacturers and/or landlords to use the newly proposed cooling techniques with minimum acceptable changes in building construction materials, and overall design. Radical changes in the popular type of residential design will not be accepted by either manufacturers, landlords, or users, due to the inconvenience resulting from different building configurations or from changes in the popular living

CHAPTER II

THE AVAILABLE PASSIVE COOLING TECHNIQUES

A literature review will reveal that there are numerous passive cooling techniques currently being presented in response to different climatic needs. Most of them, however, are not applicable to hot, humid climates (such as Taiwan). It is necessary, therefore, to discuss the most popular techniques and to understand their potential and limitations.

1.0 LOAD CONTROL

Load control means the minimization of intruding cooling loads through the weather walls from external weather conditions. Attention to this issue is always desirable and useful regardless of whether the building is a passive system or not. The four most influential factors in load control will be discussed as follows:

1.1 Orientation

The best orientation for passively cooled buildings will assure a minimization of direct solar radiation and will take full advantage of the prevailing wind patterns. B. Givoni¹ pointed out building orientation affects the indoor climate in two respects:

- a) Solar radiation and its heating effect on walls and rooms facing different directions.
- b) Ventilation problems associated with the relation between the direction of the prevailing winds and the orientation of the building.

A time-proven and general rule of thumb is to orient the building toward true south, with its main facades, and largest external wall areas facing south and north. Its minor facades will therefore be exposed to the east and west (Fig. 2.1).

This orientation is best because the low summer sun in the east and west creates excessive solar radiation on the east and west facing walls or windows due to the incident beam radiation. These rays strike the vertical facades at low angles on the east and west side, as compared to the steeper angles of incidence on the south side (Fig. 2.2). Therefore, conduction heat gain through the walls, or radiation heat gain through the windows will increase dramatically to the east and west, compared to the south and north.

Another more recently realized reason for true south orientation is that shading is easier to accomplish where the window actually faces true south. The variation in the profile angle (the angle between the normal to window and the rays of the sun projected in a plane perpendicular to the window plane) of the sun is minimized for south facing

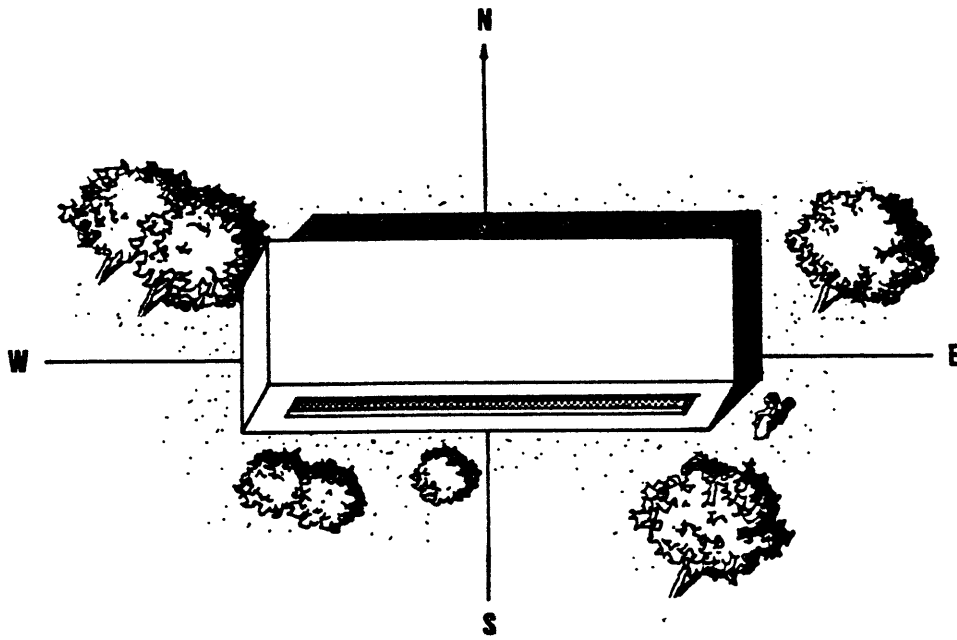


Fig. 2.1 General Rule-of-Thumb for Building Orientation

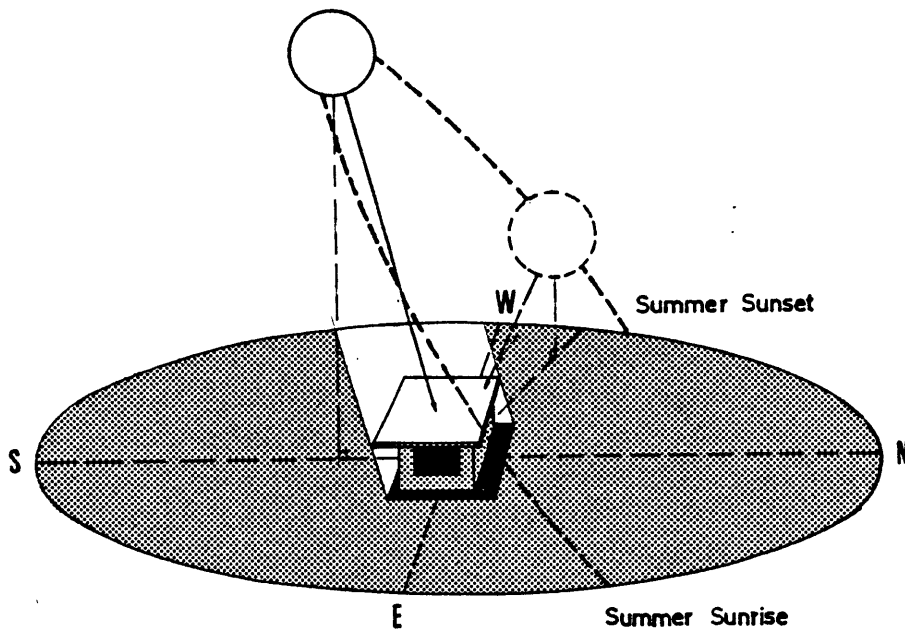


Fig. 2.2 Solar Incidence on the South and West Facades

windows (Fig. 2.3). A good south facing horizontal shading device can effectively shade the summer sun, and still let in the winter sun.

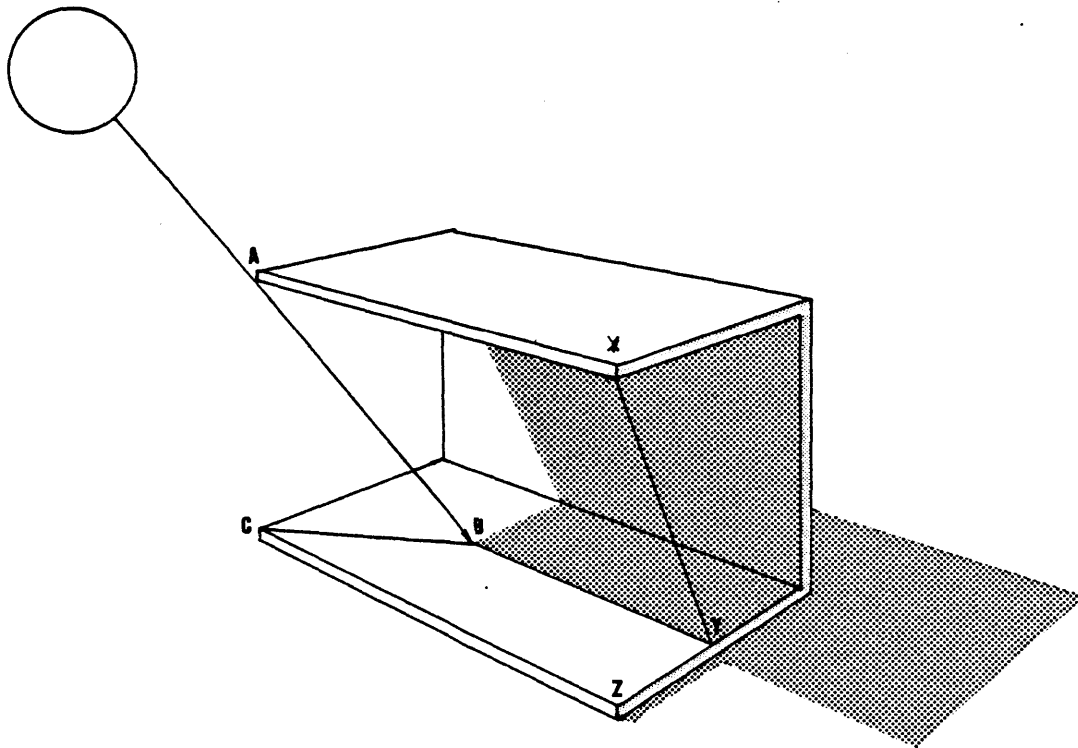


Fig. 2.3 Profile Angle $\angle XYZ$

1.2 Window and Sunshading Design

Solar radiation through the windows is obviously to be avoided in hot, humid climates. This task can be accomplished by using horizontal overhangs and vertical fins. Shading coefficients for different interior or exterior venetian blinds, roller shades, and curtains (Olgyay², "Design with Climate"), can also be applied to

solar heat gain factors to estimate the resultant reduction in gains.

Horizontal overhangs are useful to control the sun on the south side because the altitude of sun's position is high to the south. The vertical fins are useful to control the sun on the east, west, or north side because the altitude of the sun is low at these orientations.

Reducing window areas is very efficient in solar radiation control. This can be difficult, however, because large window areas may be required for optimal daylighting and natural ventilation. Nevertheless, it is always good to reduce the east and west side window areas, and increase those on the north or south, so that the same total window area, and daylight distribution inside will result.

Chapter III will discuss sun chart and shading mask protractor. With these tools one can appropriately design shading devices for different orientations.

1.3 Choice of Materials and Colors

The choice of building materials and colors holds significant importance in the reduction of solar heat gains through opaque walls and roof.

In hot, humid climates, low-mass construction is preferable for residential buildings because the overheating problem only occurs during the daytime when most of the family has gone out for work, school, shopping, etc. High-mass construction is not applicable to hot, humid

climates because high air temperatures (usually greater than 78 °F) and high humidities at night make it impossible to cool the mass enough to allow it to serve as a heat sink during the daytime. The heat absorbed in the daytime, therefore, is released from the mass during the night time, when most of the family is at home. The building would then still need cooling rather than heating. In a region where reinforced concrete construction has dominated, by reason of its durability, the easiest way to reduce the concrete mass (heat storage) is to cover the floor with carpets.

Insulation within the walls and roofs can greatly reduce instantaneous solar heat gain. Care must be taken, however, in a space without air-conditioning because although adding more insulation will appreciably reduce the day time temperature, it may also result in a high internal surface temperature at night (Givoni³, "Man, Climate and Architecture"). The R-value of the walls and roof must be established in conjunction with the overall passive cooling techniques to be employed.

Color is the most effective and cheapest of the load control techniques. White or other light exterior colors can reflect 80%-90% of the potential solar radiation due to their low absorptivities at shortwavelengths. These colors are also good radiators because all the colors (whether white or black) have very high emissivities (0.9) at long wavelengths. Therefore, light colors not only reflect most of solar radiation, but they also radiate energy to other

cooler objects and to the night sky. Givoni⁴ has shown significant load reductions through the use of light-colored roofs and walls.

1.4 Ventilation Control

In hot, humid climates ventilation may be a significant source of heat gains (sensible and latent heat) due to the high air temperatures and high humidities which are brought into the room by the winds. The ventilation schedule must be constructed according to the accessible local weather data, which shows hourly air temperature and relative humidity readings. When the building is not to be ventilated, sealed joints at the windows and door frames of the weather wall must be used to reduce the infiltration heat gains.

2.0 CONVECTIVE COOLING

2.1 Physical Effect

Air movement creates a cooling sensation not by decreasing the ambient air temperature but by reducing the skin temperature, due to heat loss by convection and due to the increased evaporation of moisture (sweat). Theoretically, the higher the air velocity, the greater cooling effect that can be obtained due to increased convection and evaporation. Table 2.1 taken from Olgyay⁵ shows the desirable range of wind velocities which are compatible with human comfort.

VELOCITY	PROBABLE IMPACT
Up to 50 fpm	- Unnoticed
50 to 100	- Pleasant
100 to 200	- Generally pleasant but causing constant awareness of air movement
200 to 300	- From slightly drafty to annoyingly drafty
Above 300	- requires corrective measures if work and health are to be kept in high efficiency

Table 2-1

In hot, humid climates, however, an increased air velocity may be quite ineffective as a cooling medium, because high ambient air temperatures (above 90 F) reduce the convection heat loss from the body and high moisture content in the air suppresses its ability to pick up additional moisture from the surface of the skin. At the same time, the hot, humid ventilation air moving through the building not only increases the structural temperature, but is also prone to causing mold and mildew problems.

2.2 Natural Ventilation

Natural ventilation is a time proven cooling concept. The effectiveness of this technique depends on the total effects of air movement, temperature, and humidity. The most general method to evaluate its effectiveness is to plot the monthly average temperature and humidity on a psychrometric or a bioclimatic chart (see Chapter III). This procedure indicates the periods during which the temperature and humidity are such that ventilation could be used to accomplish the required cooling effect. Indications of the wind velocity needed during these periods can be obtained from the bioclimatic chart. During the spring and fall when temperatures are moderate, ventilation is generally an effective cooling strategy.

Givoni⁶ has shown that the inlet windows do not necessarily need to face perpendicular to the prevailing winds. It may, in fact, be better to have the winds oblique to the inlet opening because in this case air becomes turbulent in the room, thereby increasing the air flow along the side walls and in the corners, causing a more even distribution of the cooling effect. When fly-screens are applied directly to the windows, however, the reduction in internal air velocity is greater with an oblique than with a perpendicular wind. This is because the oblique wind slides over the screen and does not create an efficient pressure in front of the window.

Cross-ventilation should be addressed as the most

effective type of natural ventilation. This term refers to conditions in which the given room has windows which are exposed to both pressure and suction areas of the wind. The average air velocity in a cross-ventilated room is more than double that in rooms with window openings only toward the pressure area, or only toward the suction area of the external winds.

Regarding the window's position in the section of the room, Olgyay⁷ has made the conclusion that the location of inlet opening plays the major role in determining the internal air flow pattern while the location of the outlet has little effect on it. A low inlet causes a downward air flow pattern and can effectively cool the occupants of the room.

2.3 Thermally Induced Ventilation

This type of ventilation is also called the "chimney effect" because it has same effect as the draft caused in the flue of a fireplace. The basic concept is that when air is heated in a vertical column by solar energy (or other heat source), it rises due to its lower density per unit volume. Ambient unheated air is pulled in to replace the rising air leaving through the "chimney". Unlike natural ventilation, this system does not require a wind flow or any particular building orientation.

Thermally induced ventilation, however, has several serious limitations. Abrams and Brock⁸ have shown that wind

velocities of 3-5 mph are sufficient to entirely overpower the "chimney effects". These wind velocities can usually be obtained in most of the regions on earth. In addition, Akridge⁹ pointed out a more serious problem resulting from the fact that solar energy is most available during those times when the ambient air temperatures are the highest. Therefore, a solar chimney works best during those times when people would prefer to keep the extra heat outside the building rather than introducing more to stimulate the air flow.

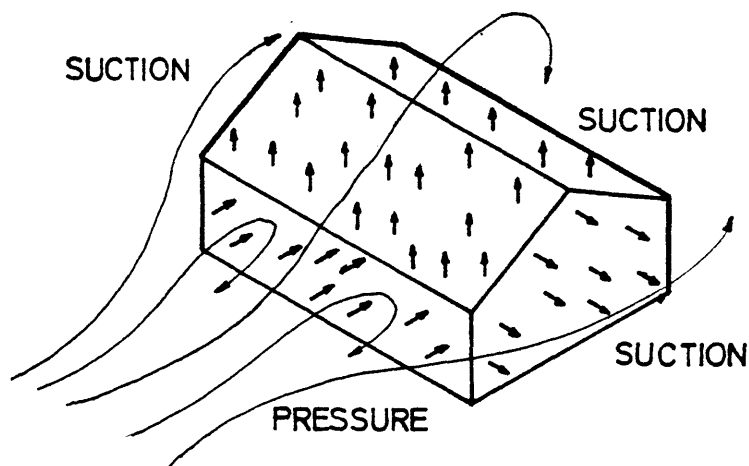


Fig. 2.4 Typical Wind Pressure Distribution on A House

2.4 Wind Induced Ventilation

When wind blows towards a structure shown as Figure 2.4 the windward side of the wall experiences a positive pressure which pushes the air against the wall. The leeward side, roof, and the walls experience a negative pressure which creates a suction on the surface.

The pressures are turbulent due to the fluctuating nature of the wind. The negative pressure on the roof and the sides are nonuniform, with the maximum effect of the suction around the corners and the edges. The positive pressure pushes the air into the structure through the openings while the negative pressure pulls the air out of the structure through the openings. These two effects work together in wind induced ventilation.

Wind induced ventilation may be applied when natural ventilation is ineffective for the following reasons:

- 1) The wind velocity is low.
- 2) The openings only face one pressure zone (either positive or negative zone).
- 3) The limitation of openings for physical or security reasons.

When wind velocity is low, many authors believe that the size of the outlet to inlet opening ratio is optimized at 3 :1 in order to create high pressure difference at inlet opening and as such increase the interior wind velocity.

This suggestion, however, is somewhat misleading. According to the research of Harris Sovin¹⁰ in Arizona, the air velocity is increased but only adjacent to the inlet opening. The average air velocity is maximum when outlet-inlet ratio in window size is 1.25:1. The use of mechanically driven ventilation can be much more effective when the prevailing wind velocity is fairly low.

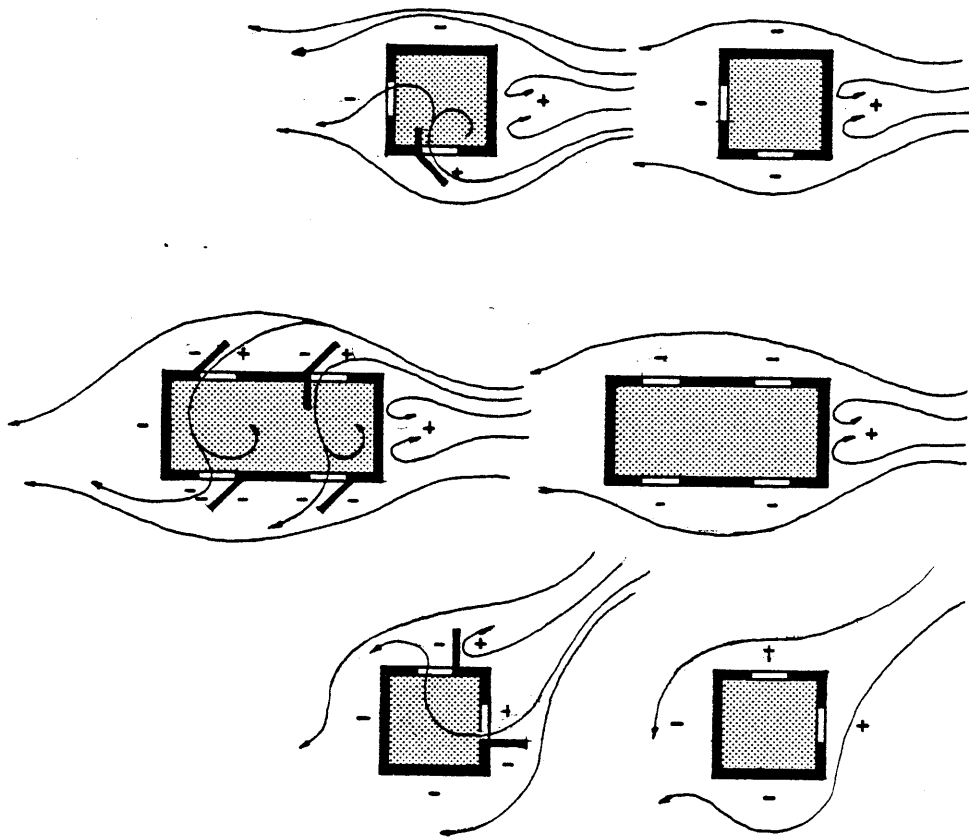


Fig. 2.5 The Effect of Applying Wind-Inducing Devices

If the openings in the building only face one pressure zone, either positive or negative zone, a technique to increase natural ventilation is to use wind inducing panels to change the pressure pattern. Figure 2.5 shows some examples of this application in plan.

Where there is a limitation of openings for physical or security reasons, wind scoops are an alternative type of wind inducing form. Many desert areas are subject to constant prevailing winds. In these areas the winds near the ground are sandy, and dense settlements are necessary due to sparse water resources and defense requirements. Many wind scoops were therefore built to address these needs, and to catch and funnel prevailing winds into the structures.

2.5 Night Ventilation

When the ambient air is cool at night, ventilation can be used to cool the structure or the occupants. This technique has considerable merit in hot, dry climates where the daily temperature range is greater than 30°F. The mass of the structure is cooled by the lower temperature of the night air. When the temperatures begin to rise the next morning, the structure is closed up and remains so until the following night.

In hot, humid climates, however, due to daily temperature amplitudes which are low (on the order of 15°F), the night air temperature is not low enough (around 78°F) to cool the structure. Night ventilation in this case may only

be capable of cooling the occupants directly at night. Since the most effectively sealed building has at least one quarter its volume of air changing (due to infiltration) each hour, night ventilation does not notably increase the latent load (moisture content) in the building, but it does decrease the sensible load (causing a sensible cooling effect by convection from human body).

2.6 Mechanically Driven Ventilation

Although mechanically driven ventilation cannot strictly be grouped into passive cooling techniques because it uses electrical energy, it may be called "passive" in the case that no refrigeration is utilized. This hybrid system uses a fan, driven by electrical energy, through which air movement is created in the desired place.

The electric fan is a time proven cooling device in hot, humid climates. These systems can be divided into two categories: direct and indirect systems. A direct system blows the air directly onto the occupants, while an indirect system blows the air into the space. A direct system is preferable when intermittent usage and maximum air velocities are needed. An indirect system is desirable when constant usage, higher mechanical efficiency, and quietness are at a premium.

When prevailing wind speed is low, but air temperature and humidity conditions would allow for adequate cooling from ventilation alone, it is best to use a fan which will

only consume about 20% of the total energy involved in using even the most efficient of air-conditioning systems.

3.0 EVAPORATIVE COOLING

Evaporative cooling has shown its significant potential in hot, dry climates. The principle is that when water evaporates, it cools the ambient air by capturing the heat necessary for evaporation. This technique, unfortunately, has no beneficial effect in hot, humid climates, because the moisture content here is already higher than that which is comfortable, and further evaporation would result only in increased discomfort.

Although this technique cannot be applied in hot, humid climates for ventilation air, it can be used with roof sprayers to reduce the tremendous sensible cooling load through the roof. Roof spraying has been proven, by Bacon,¹¹ to be a very effective cooling system in terms of reducing roof temperatures by as much as 60°F through the use of it.

4.0 RADIANT ROOF COOLING

This technique uses the roof as a large radiator to dissipate building heat to the cold ambient air and to the dark, clear, night sky. For normal roof materials, radiant cooling is only effective when the mean relative humidity is lower than 60%. Evidently, then, radiant roof cooling can have little positive effect in hot, humid climates where mean relative humidity is always 70%-80%.

5.0 EARTH COUPLED COOLING

Soil temperatures are usually cooler in summer and warmer in winter than the associated air temperatures. This technique uses soil as insulation and as thermal mass to moderate the indoor-outdoor temperature differentials. This type of system usually requires that the houses be buried or at least semiburied. Although much attention has been given to this type of construction, the problem of condensation on the walls (due to high humidity), and the resultant lack of natural ventilation and daylighting, together with high mean soil temperature, and high land prices (due to high population density in some developing countries), are all forces that work against the effectiveness of this technique in hot, humid climates.

6.0 DETACHED EARTH COUPLED COOLING

The structures using this cooling technique are built above the ground, as opposed to those using earth coupled cooling. This technique can be divided into two different categories in terms of the employed energy transfer medium (air or water). Both work on the principle that as the medium is circulated through the pipes or tubes buried in the ground, the temperature of the medium becomes cooler as the ground temperature is cooler than the air temperature at 6-8 ft depth in summer. As the air or water is discharged from the ground tubes or pipes, it can be utilized to cool

the occupants or the structure.

Akridge¹² pointed out that the type of system which uses air as a medium shows serious problems. This type usually uses the cooled air directly to cool the occupants. Due to the high humidity in hot, humid climates, the pipes usually only cool the air to the dew point temperature and thus do not remove any latent load. The discharged cool air is nearly saturated, (100% relative humidity) and it is consequently quite uncomfortable.

Even if the pipes are long enough to cool the air to produce condensation moisture in the pipes is difficult to dispose of, and is therefore prone to producing mold, mildew, and bacteria which may creat health or odor problems.

If water is used as the heat transfer medium, the system is usually used to cool the mass rather than the occupants directly. Here the occupants are cooled by radiant coupling to the mass. This is a closed-loop system which pumps the water through pipes in the ceiling (or the walls) and then the ground. Akridge¹³ has demonstrated that this type of detached earth-coupled cooling has great potential in reducing sensible load in hot, humid climates.

The mean ground temperature is a decisive factor in the success of this technique. However, in some hot, humid climates when mean ground temperatures are above 72°F, the use of this technique is evidently of dubious value since the water must be 7-8°F cooler than the desired room air

temperature when using cooled interior masses.

7.0 DEHUMIDIFICATION

All of the passive cooling techniques mentioned above are applied to reduce the sensible loads, i.e., to reduce ambient air temperatures. In hot, humid climates, even when the temperature is in the comfort range (68-78 °F), uncomfortable conditions may still occur due to high vapor pressure in the air. This excessive vapor pressure, caused by high humidity, is called the latent load (because it is concealed) and it must also be reduced to restore the feeling of comfort.

Various chemical desiccants can be used to remove moisture, thus reducing the latent load. Unfortunately, the heat released from the liquification of water vapor and its absorption by desiccants significantly raises the air temperature. That is to say, the latent load is decreased while the sensible load is increased. Fig. 2.6 shows a psychrometric chart analysis of typical desiccant dehumidification performance. This chart reveals that, as air is dried by desiccants, the dry-bulb temperature of the air is raised dramatically.

Some passive solar desiccant air-conditioning systems have been proposed, but their commercialization is quite dubious due to their extremely high cost.

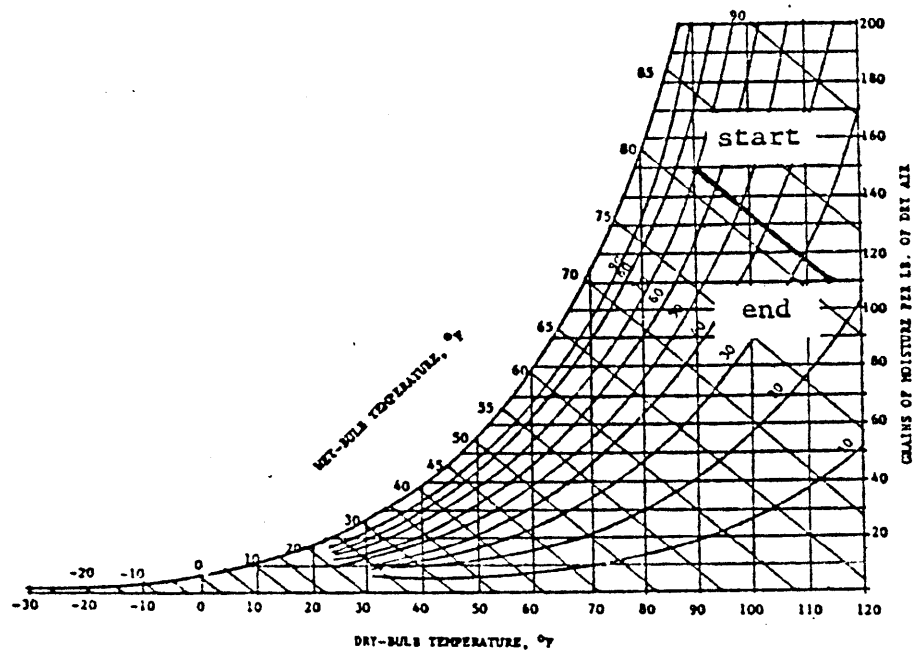


Fig. 2.6 Sorbent Dehumidification on Psychrometric Chart

REFERENCES FOR CHAPTER II

1. B., Givoni, "Man, Climate & Architecture", (Applied Science Publishers Ltd., 1976)
2. V., Olgyay, "Design with Climate", (Princeton University Press, 1963); pp.67-71.
3. B., Givoni, "Man, Climate & Architecture", (Applied Science Publishers Ltd., 1976), chapter 7.
4. Ibid.
5. V., Olgyay, "Design with Climate", (Princeton University Press, 1963).
6. B., Givoni, "Man, Climate & Architecture", (Applied Science Publishers Ltd., 1976), chapter 15.
7. V., Olgyay, "Design with Climate", (Princeton University Press, 1963), chapter 9.
8. J.M., Akridge, "Investigation of Passive Cooling Techniques for Hot-Humid Climates", (College of Architecture, Georgia Institute of Technology, October 1982), Chapter 2.
9. Ibid
10. G., Wittenberg and W., Turner, "Climate Based Passive Cooling", Teaching Passive Design in Architecture, (Association of Collegiate Schools of Architecture, November 1981), section 4.
11. J.M., Akridge, "Investigation of Passive Cooling Techniques for Hot-Humid Climates", (College of Architecture, Georgia Institute of Technology, October 1982).
12. Ibid.
13. Ibid.

CHAPTER III

WEATHER DATA AND DESIGN TOOLS

In order to design a climate-responsive building, especially with regard to its passive cooling potential, architects must know what types of weather data are essential, and how they may be analyzed. This chapter will focus on weather data, and their impact on design tools. The weather data of Taipei, Taiwan will be used as an example to illustrate the analytical method being developed.

1.0 WEATHER DATA

1.1 Geographical Position

The geographical position of a region determines the sunpath, magnetic deviation, and the time lag between solar time and local time. As a result, knowing one's position on the globe is very important for sunshading design. It also suggests the large regional climatic influences. The geographic position is defined by 3 parameters: latitude, longitude, and elevation. For Taipei these parameters are $25^{\circ}04$ North, $121^{\circ}33$ East, and 20 ft respectively.

In addition to regional climatic influences, each individual building site has its own specific set of microclimatic influences. For example, the sun or prevailing winds may be obstructed by adjacent high-rise buildings. In urban areas, the microclimatic relationships

become quite complex and often vary from the more general and larger regional influences. These relationships, however, must be individually examined at the specific site and as such, they are beyond the scope of this thesis. This thesis intends to suggest general design solutions based on regional statistics for Taipei.

1.2 Essential Weather Data

Olgyay pointed out that the major elements of the climatic environment which affect human comfort can be categorized as follows: 1) air temperature, 2) humidity, 3) air movement, and 4) radiation.

1.2.1 Air Temperature

Air temperature data are important primarily in comparison to the desired interior air temperatures for determining the heating and cooling loads. Temperature as measured by a standard thermometer is called the dry-bulb temperature. Temperature as measured by a "special" thermometer whose bulb is covered with a wet wick and exposed to moving air is called the wet-bulb temperature. Knowing these two temperatures, the moisture content of the air or the relative humidity can be found, and from this information, the dew-point temperature, or that temperature at which the moisture of the air begins to condense out of the air as it cools, can be derived from a psychrometric chart. The vapor pressure in the air, which influences

human comfort, and potential condensation problems in building materials can then be examined, and properly evaluated.

1.2.2 Humidity

Humidity, as mentioned above, refers to the moisture content in the air. absolute humidity is defined as the weight of water vapor per unit volume of dry air. Relative humidity (R.H.) is defined as the ratio of the actual absolute humidity to the maximum possible moisture content of dry air at a given temperature.

The same R.H. does not mean the same moisture content in the air at all times, because air of a higher temperature can hold more moisture. For example, at a given absolute humidity, air of a low temperature will have a higher R.H. value than air of a higher temperature. Therefore, R.H. has no meaning as an environmental index unless it is correlated with an accompanying dry-bulb temperature.

A high absolute humidity will make summer temperatures seem hotter by reducing the evaporation rate from the surface of the skin. The vapor pressure in the air is always higher outside than inside conditioned space in summer. This difference is used to calculate the latent cooling load (see Appendix A).

1.2.3 Air Movement

Wind is a major cooling influence. Wind direction data

are most helpful in determining building orientation, opening locations, and wind-inducing devices which direct the wind. Wind velocity data are essential for determining whether the wind intensity is sufficient to cool the people inside a building.

1.2.4 Solar Radiation

Solar radiation data indicate the availability of solar energy. The data can be used to determine the radiation heat gain through the fenestration and the conduction heat gain through the opaque walls or roof for the purpose of cooling load evaluation.

Since these data have not yet been established by the central weather bureau, R.O.C., a table of clear-sky solar heat gain factors for 24 Deg. North Latitude in ASHRAE Fundamentals is used in this thesis in calculating cooling loads (see Appendix A).

The west facing facades are shaded in the morning and only experience the diffuse component of solar radiation in this table. These half-day heat gain factors through a 0.125 inch sheet glass are used to calculate the average cooling load. Because the building proposed in this thesis is totally shaded in summer, it is only exposed to the diffuse component of solar radiation. These factors show their greater accuracy due to the high cloudiness and precipitation rate of Taipei where diffuse radiation is dominant.

1.2.5 Other Helpful Data

In addition to the essential regional weather data mentioned above, there are some other data which are helpful in passive cooling design.

Relative clearness (due to clouds or air pollution) and precipitation rates are helpful in predicting the pattern of solar radiation (diffuse or direct), when measured radiation data is unavailable. Soil temperatures at different depths are helpful in predicting the feasibility of the earth coupled or detached earth coupled cooling techniques. Water table level information is important for determining the heat flux impact on the buried cooling reservoir which is proposed in Chapter V. Extraordinary or rare phenomena such as typhoons must be taken into account in designing wind-inducing or sunshading devices, which must be able to withstand extremely high wind pressures (19.2 m/sec).

1.3 Hourly Weather Data

Hourly weather data are most essential because they show the dynamic changes during each day throughout a typical year. They are usually recorded as averages of each hour over a 10 year period. These data are, however, too numerous to analyze without the aid of a computer. Therefore, they are not very useful to architects who want less cumbersome guides to the estimation of heating or cooling loads in buildings.

As more microcomputer programs are developed for weather data analysis, the potential of using hourly weather data for a climate responsive design will increase. An example of hourly weather data of one day for Taipei is shown in Table 3.1.

1.4 Monthly Average Hourly Weather Data (M.A.H.W.D.)

M.A.H.W.D. are derived from the average of hourly weather data for each month. Although M.A.H.W.D. only represent one typical day for each month, they are much easier for architects to analyze and manipulate. With knowledge of the cooling load calculation methods developed by ASHRAE, we can quickly estimate the cooling load for the given typical day, and thereby derive an estimate of the loads for the whole month, by simply multiplying the total daily cooling load by the number of days in that month. The monthly average peaks and lows for outdoor air temperature given by M.A.H.W.D. can also be used to calculate the peak heating or cooling loads.

In addition, it is possible to compare the effect or relative benefit of various design solutions through the application of these averages. Once the benefits and costs of each solution are known, it becomes easier to choose those proposed solutions which are both efficient and economical.

Even if the computerized weather analysis is available, it is still very useful for architects to be familiar with

JOB OF CENTRAL WEATHER BUREAU (R.O.C)											
METEOROLOGICAL DATA OF TAIPEI											
MONTH: 1 DAY: 31											
HOURLY	AVERAGE	METEOROLOGICAL	DATA	OF	TAIPEI	MONTH: 1	DAY: 31				
HOUR	DBT	DEW-POINT	PRESSURE	CLOUD	RAINS	W-SPD	RH	T-10	T-20	T-30	SUN
1	13.5	11.7	1019.3	3	0.0	3.6	88				0.0
2	13.5	11.5	1019.3	8	0.1	3.2	88				0.0
3	13.3	11.5	1019.1	4	0.0	2.5	89				0.0
4	13.2	11.5	1019.0	4	0.2	2.6	90				0.0
5	12.9	11.2	1019.2	2	0.3	2.4	89	16.1	17.0	17.3	0.0
6	12.8	11.0	1019.7	4	0.3	2.6	89				0.0
7	12.6	11.1	1020.2	4	0.4	2.3	90				0.0
8	12.7	11.0	1020.5	8	0.4	2.5	89				0.0
9	13.7	11.5	1020.9	7	0.4	2.3	86				0.2
10	14.8	11.9	1020.5	4	0.3	3.1	83				0.3
11	15.9	12.0	1019.5	7	0.3	3.3	78				0.3
12	16.6	12.6	1019.0	4	0.2	2.7	78				0.3
13	16.7	12.4	1018.2	4	0.3	3.3	76				0.3
14	15.9	12.2	1017.7	8	0.3	3.3	79	16.5	16.9	17.2	0.2
15	15.8	12.4	1017.7	4	0.3	2.7	80				0.2
16	15.4	12.0	1017.9	5	0.2	2.5	81				0.1
17	15.2	11.9	1018.3	9	0.1	2.3	81				0.0
18	14.8	11.9	1018.8	5	0.3	2.2	83				0.0
19	14.5	12.0	1019.2	5	0.2	2.0	84				0.0
20	14.3	12.2	1019.3	9	0.3	2.1	87				0.0
21	14.0	12.1	1019.5	9	0.2	1.7	88	16.4	17.0	17.2	0.0
22	13.9	11.9	1019.6	4	0.2	1.8	87				0.0
23	13.8	11.7	1019.4	8	0.4	1.6	87				0.0
24	13.6	11.6	1019.3	3	0.3	1.4	88				0.0

TABLE 3.1

DAILY AVERAGE: MONTH OF AUGUST											
DBT DEW-POINT WIND SPEED											
DEG.CEL.	DEG.FAR.	DEG.CEL.	DEG.FAR.	M/SEC	NPH	RH	SUN				
1	28.0167	82.4300	23.8625	32.0000	3.0208	6.7576	79.0833	0.1958			
2	28.6833	83.6300	23.5500	32.0000	2.7667	6.1890	75.0417	0.2708			
3	28.4583	83.2250	23.9042	32.0000	2.7625	6.1797	77.0000	0.2083			
4	28.3083	82.9550	24.0958	32.0000	2.0208	4.5206	78.5833	0.2417			
5	28.3458	83.0225	23.6542	32.0000	2.2917	5.1265	76.5417	0.2708			
6	28.8833	83.9900	23.2167	32.0000	3.0667	6.8601	72.8333	0.2542			
7	28.7083	83.6750	23.8042	32.0000	2.7333	6.1145	75.4583	0.2583			
8	29.1000	84.3800	23.5917	32.0000	3.0417	6.8042	72.8750	0.3125			
9	28.8333	83.9000	23.6167	32.0000	3.3542	7.5033	74.3750	0.2000			
10	28.6375	83.5475	23.5167	32.0000	3.4500	7.7176	74.6250	0.1833			
11	28.5917	83.4650	23.7000	32.0000	2.8417	6.3568	75.7083	0.2458			
12	28.4792	83.2625	23.6417	32.0000	2.4958	5.5832	76.0833	0.1750			
13	28.5083	83.3150	23.7250	32.0000	2.6375	5.9001	76.2500	0.2125			
14	29.0292	84.2525	23.6333	32.0000	2.8167	6.3009	73.7917	0.2750			
15	28.8417	83.9150	23.6792	32.0000	3.2083	7.1770	74.8750	0.2875			
16	29.3875	84.8975	23.7042	32.0000	3.1292	6.9999	72.2500	0.2417			
17	28.7458	83.7425	23.8125	32.0000	3.0458	6.8135	75.7083	0.2292			
18	28.9667	84.1400	23.7292	32.0000	2.2208	4.9680	74.3333	0.2750			
19	28.8750	83.9750	23.8167	32.0000	2.4167	5.4061	75.1250	0.2125			
20	28.8292	83.8925	24.0208	32.0000	2.7792	6.2170	76.2083	0.1583			
21	29.0375	84.2675	23.9917	32.0000	2.6125	5.8442	74.8750	0.2083			
22	28.9792	84.1625	23.6167	32.0000	3.3500	7.4939	73.7500	0.2500			
23	29.1583	84.4850	23.3708	32.0000	4.3083	9.6377	71.8750	0.2792			
24	28.4500	83.2100	23.4792	32.0000	3.3583	7.5126	75.1667	0.1583			
25	28.4750	83.2550	22.9000	32.0000	2.7250	6.0958	72.6667	0.2708			
26	28.2042	82.7675	23.2667	32.0000	2.9375	6.5712	75.1250	0.2000			
27	27.6625	81.7925	23.3500	32.0000	2.5458	5.6950	78.1667	0.1458			
28	27.6583	81.7850	23.3167	32.0000	2.3375	5.2290	77.8333	0.1917			
29	27.9125	82.2425	23.1958	32.0000	2.1750	4.8655	76.1250	0.2875			
30	27.8125	82.0625	22.9333	32.0000	2.2000	4.9214	75.4167	0.2750			
31	27.5292	81.5525	22.8583	32.0000	2.4125	5.3968	76.3333	0.2125			

TABLE 3.2

M.A.H.W.D. analysis, especially during the preliminary stages of design, in order to understand the assumptions made in computer programs. Assumptions made in these programs can cause limitations in the analysis which are difficult to recognize unless the architect has experience in M.A.H.W.D. analysis. Table 3.2 shows the M.A.H.W.D. of August for Taipei.

2.0 DESIGN TOOLS

2.1 Psychrometric Chart Analysis

The psychrometric chart is a graphic representation of air temperature, humidity, and other data that are important in climate responsive design. (Fig. 3.1) The dry-bulb temperature (shown by the curved lines that rise from left to right) are used as a basis to determine appropriate climate responsive design strategies.

Milne and Givoni proposed a method to analyze the weather conditions. They divided the psychrometric chart into several "zones" (Fig. 3.2). The zone which lies in the center of the chart is named the "comfort zone". This comfort zone, based on the average American attitude, falls roughly between 68° F to 78° F and 20% to 80% R.H.. The comfort boundaries exclude the "hot-humid" corner of these coordinates. If weather conditions (dry-bulb temperature and the accompanying R.H.) fall outside the "comfort zone", strategies may be derived from the chart which will restore comfort conditions.

The Psychrometric Chart.

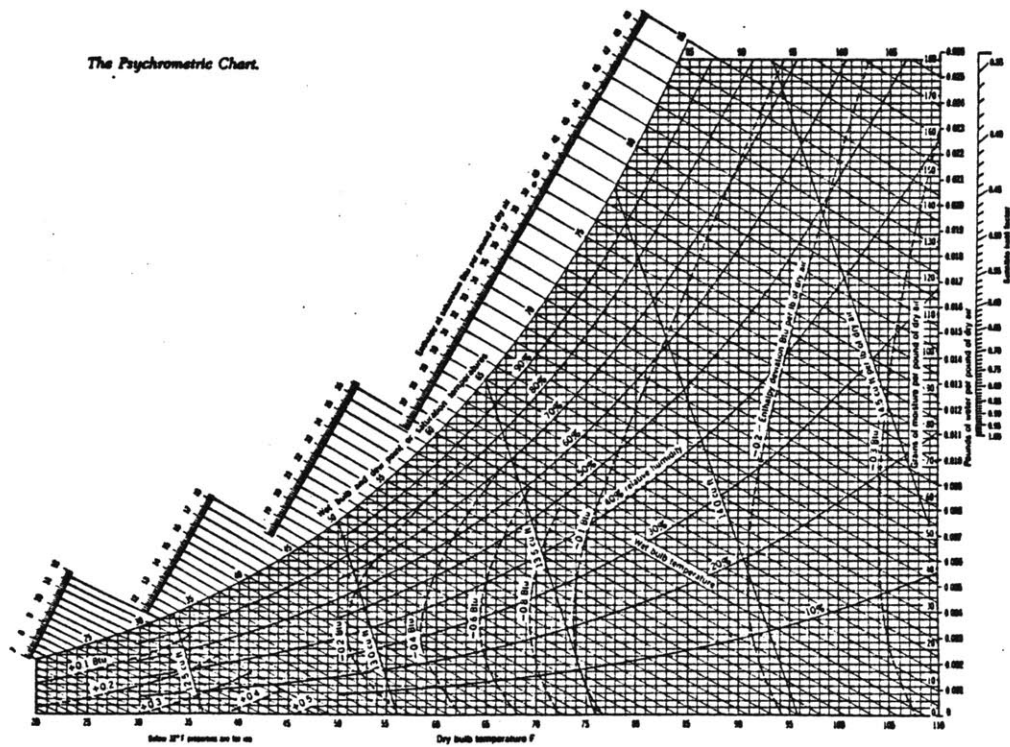


Fig. 3.1 Psychrometric Chart

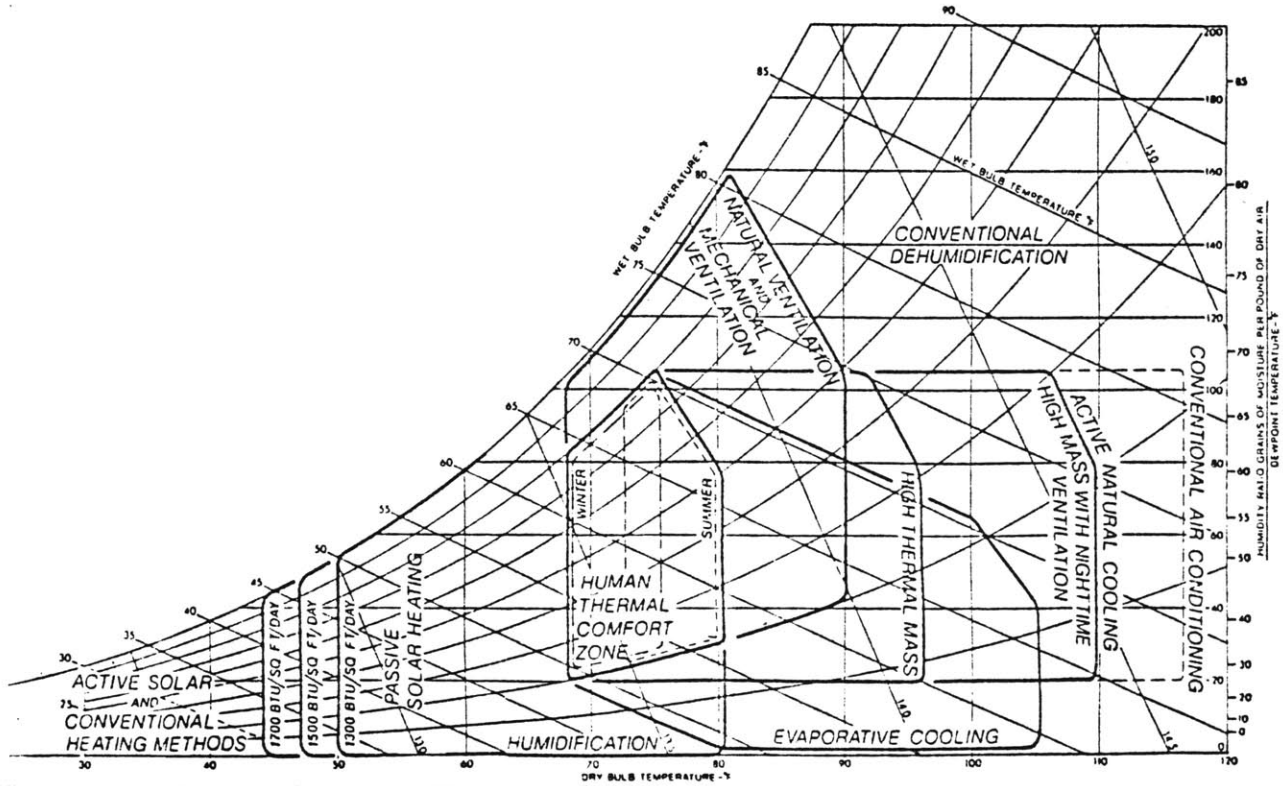


Fig. 3.2 Givoni's Psychrometric Chart

Once the M.A.H.W.D. are plotted on a psychrometric chart, we can realize which strategies are best used for each month. Fig.3.3 shows Taipei's average monthly weather conditions as plotted on the psychrometric chart. The 8 individual points which are connected to form the "climate curve" for each month, represent readings on a typical day taken every 3 hours from 1am to 10pm.

From strategies and climate curves shown in Fig. 3.3, we can conclude that passive solar heating is necessary and possible in Taipei between late November and the end of March. Furthermore, from May to October ventilation is needed to maintain comfort. During the day in July and August conventional dehumidification becomes necessary. Only for November and April do the daytime conditions fall naturally within the comfort zone.

2.2 Bioclimatic Chart Analysis

Bioclimatic Chart and its analytical method were first developed by Olgyay in 1963. The chart (Fig. 3.4) was constructed with dry-bulb temperature as the vertical axis and R.H. as the horizontal axis. Any climatic condition determined by its dry-bulb temperature and R.H. can be plotted on the chart. The comfort zone lies in the center of the chart. If the conditions fall into the comfort zone, we feel comfortable in the shade. If the condition falls outside, corrective measures must be utilized.

In the zone above the comfort zone, different wind

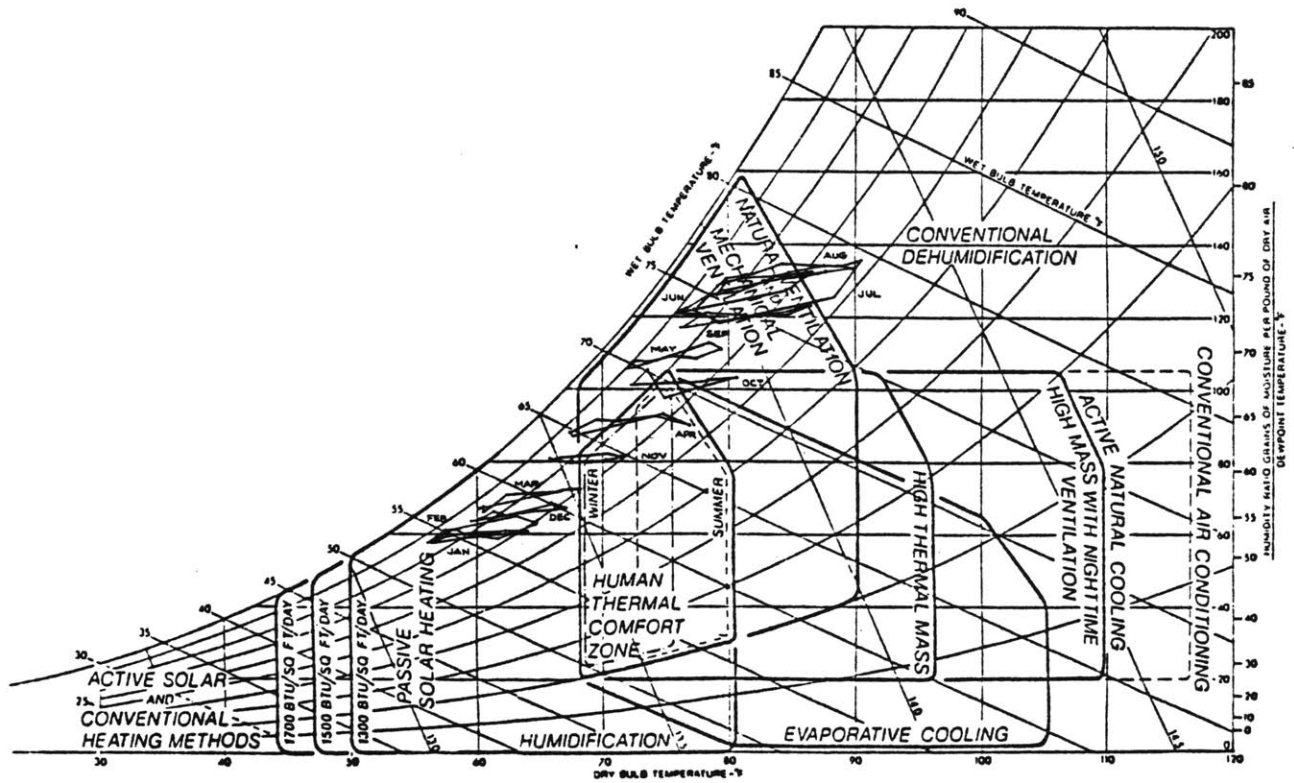


Fig. 3.3 Givoni's Psychrometric Analysis of Taipei's M.A.H.W.D

OLGYAY'S BIOCLIMATIC CHART

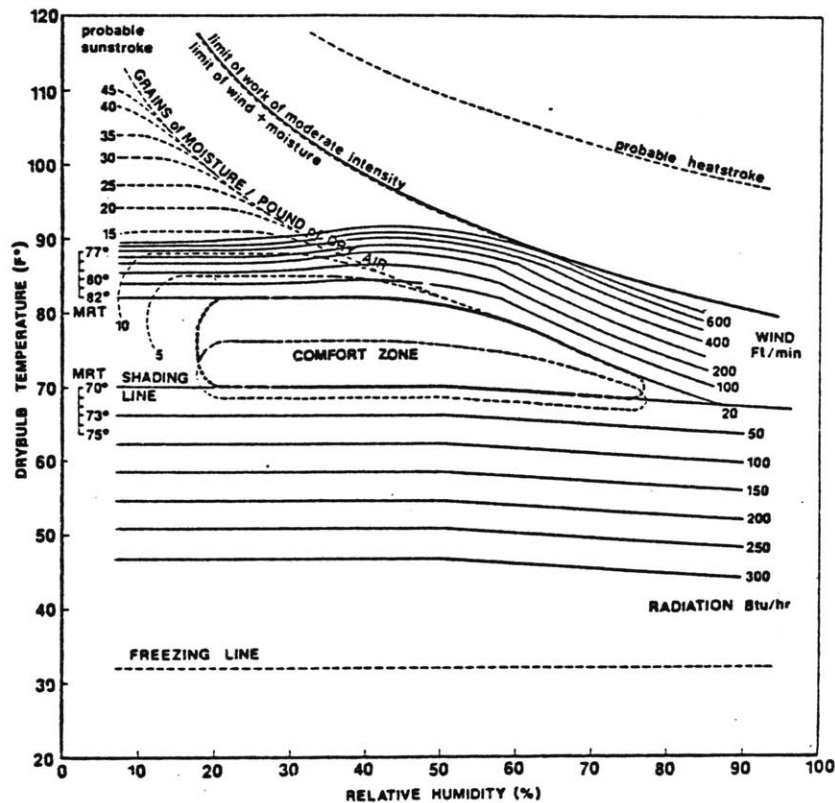


Fig. 3.4 Olgyay's Bioclimatic Chart

speeds are needed to offset the high temperatures and to restore the feeling of comfort. In the upper left zone, however, evaporative cooling is much more efficient than ventilation in restoring comfort. The zone below the shading line (above which shading is needed to exclude direct solar radiation) is divided by lines which represent the quantity of solar radiation in Btuh per square foot necessary to counteract the lower dry-bulb temperatures, and to restore the desired comfort conditions.

A new version of bioclimate chart (Fig. 3.5) was developed by Aren for the 5th International Solar Energy Conference in 1980. The major difference between Aren's and Olgyay's charts is found in the upper left zone. In this zone Olgyay proposes that moving air is of little help in cooling, while Aren proposes that it is, in fact, still very helpful.

The M.A.H.W.D. for Taipei are plotted on the Aren's bioclimatic chart in Fig. 3.6. According to this chart analysis we can conclude that from December to March, passive heating will restore comfort. Between April and December, ventilation is needed. During the day in July and August conventional dehumidification has to be utilized. Only during November do the daytime conditions fall naturally within the comfort zone.

2.3 Monthly Average Hourly Weather Data Chart (M.A.H.W.D. Chart)

AREN'S BIOCLIMATIC CHART

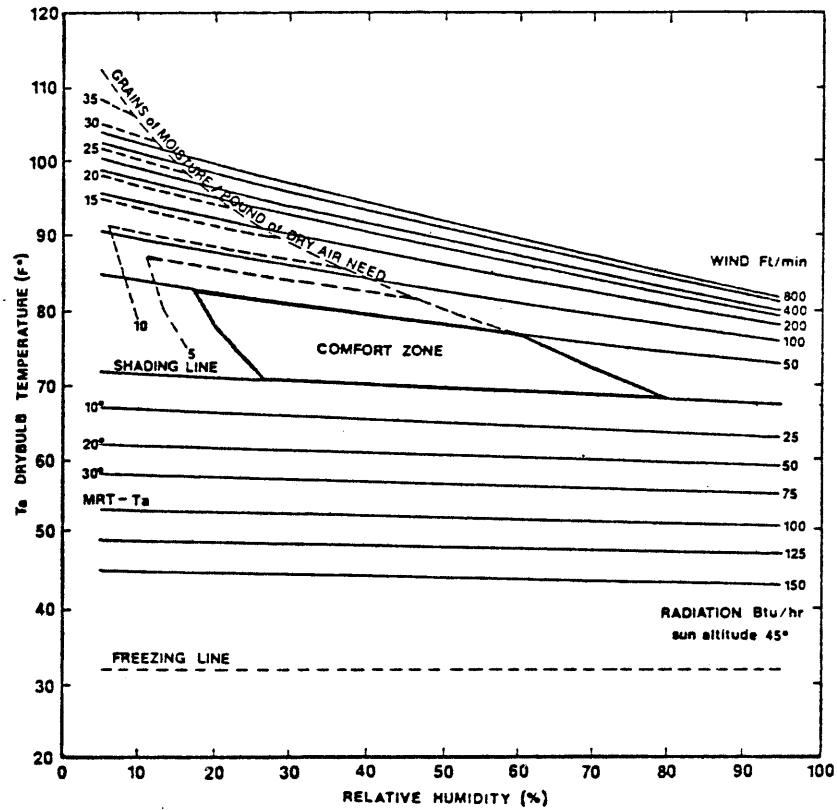


Fig. 3.5 Aren's Bioclimatic Chart

AREN'S BIOCLIMATIC CHART

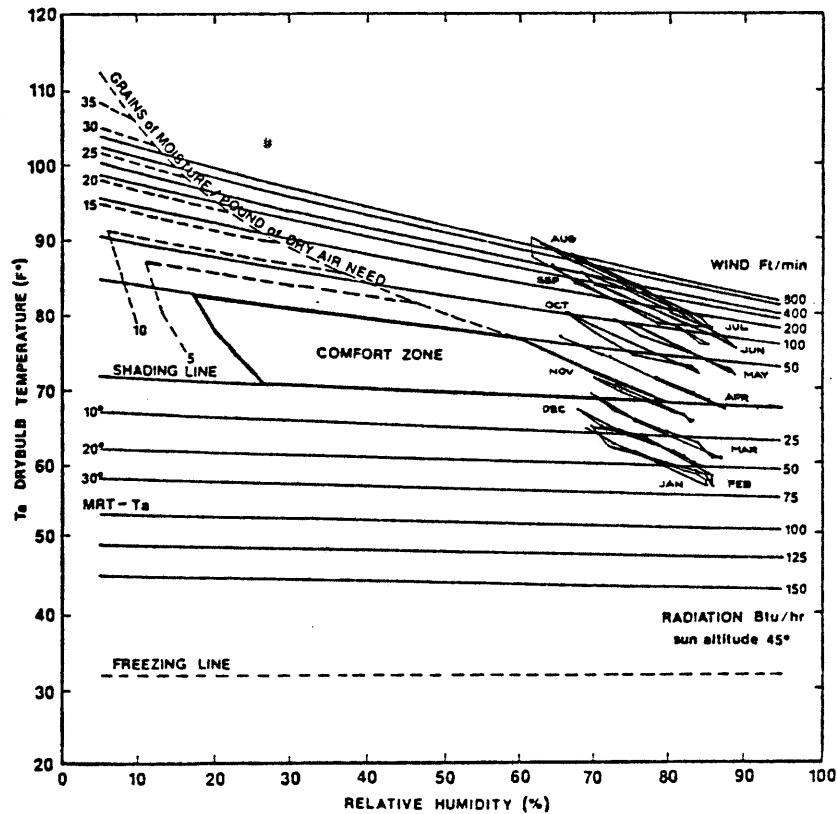


Fig. 3.6 Aren's Bioclimatic Analysis of Taipei's M.A.H.W.D

In 1980, Novell developed a simple method which uses monthly average temperatures for every other hour throughout the day as a design index to approximate the need for shading and various other potential passive cooling strategies. This method can be refined by supplying more weather data which includes hourly dry-bulb temperatures, R.H., and wind velocities to form a M.A.H.W.D. Chart. Fig. 3.7 shows this chart for Taipei, Taiwan, ROC.

Once the M.A.H.W.D. is analyzed by the psychrometric or bioclimatic charts, the different climatic needs can be transferred onto a M.A.H.W.D. chart so that the time periods for these different needs can be determined. Fig. 3.8 or Fig. 3.9 respectively show the transferred climatic needs on the M.A.H.W.D. chart analyzed by a psychrometric or a bioclimatic chart.

2.4. Sun Chart and Shading Mask Protractor

2.4.1 Sun Chart

The sun's position at any date or hour can be determined from the sun chart in terms of its altitude () and azimuth () (see Fig. 3.10). The sun chart is dependent on the geographical position and varies with different latitudes. Solar altitude is the vertical angle between the horizon and the sun's position. Solar azimuth (bearing angle) is the horizontal angle between the sun's projected position and true north. There are two popular types of sun chart: the sky-dome sun chart and the cylindrical sun chart.

Fig. 3.7 Taipei's M.A.H.W.D. Chart

MONTHLY AVERAGE HOURLY WEATHER DATA CHART										TAIPEI, TAIWAN R.O.C.			
	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	
Tmax.	64.7 RH 68.5	65.2 RH 70.1	69.3 RH 69.3	77.2 RH 65.0	80.3 RH 72.2	86.5 RH 68.0	90.6 RH 61.5	89.7 RH 63.0	86.8 RH 64.0	80.6 RH 65.7	72.0 RH 69.8	67.3 RH 67.4	
Tmin.	56.9 RH 84.8	57.2 RH 85.7	60.4 RH 87.1	67.1 RH 87.0	71.7 RH 88.8	75.6 RH 89.2	77.9 RH 89.5	78.2 RH 84.9	75.7 RH 85.0	71.6 RH 84.2	65.8 RH 89.1	59.8 RH 83.2	
Tave.	60.0 RH 78.8	60.5 RH 79.5	64.0 RH 79.5	71.4 RH 78.0	75.4 RH 81.7	80.4 RH 79.8	83.8 RH 73.8	83.4 RH 79.3	80.5 RH 75.8	75.3 RH 76.2	68.1 RH 77.7	62.8 RH 77.0	
TIME:	T RH	T RH	T RH	T RH	T RH	T RH	T RH	T RH	T RH	T RH	T RH	T RH	
DRY-BULB TEMPERATURES, R.H., & WIND SPEEDS													
0 - 1 AM	58.1 RH 83.7	58.4 RH 84.9	60.8 RH 85.7	68.4 RH 85.0	72.7 RH 87.7	76.9 RH 87.1	79.9 RH 83.9	79.9 RH 82.9	77.4 RH 83.0	72.9 RH 82.8	66.6 RH 81.9	61.0 RH 81.7	
1 - 2 AM	57.8 RH 84.3	58.1 RH 85.0	61.3 RH 86.2	68.0 RH 86.0	72.4 RH 87.8	76.5 RH 87.7	79.3 RH 84.3	79.4 RH 83.6	77.0 RH 84.0	72.5 RH 83.1	65.8 RH 82.0	60.8 RH 81.9	
2 - 3 AM	57.5 RH 84.7	57.7 RH 85.4	61.0 RH 86.4	67.6 RH 86.0	72.2 RH 88.2	76.1 RH 88.3	78.8 RH 85.0	79.0 RH 84.2	76.4 RH 84.0	72.1 RH 83.6	66.1 RH 82.3	60.4 RH 82.5	
3 - 4 AM	57.3 RH 84.7	56.7 RH 85.6	60.8 RH 86.5	67.5 RH 87.0	71.9 RH 88.5	75.8 RH 88.8	78.3 RH 85.4	78.6 RH 84.5	76.2 RH 85.0	72.0 RH 83.9	66.0 RH 82.6	60.1 RH 82.9	
4 - 5 AM	57.1 RH 84.9	57.4 RH 85.6	60.8 RH 86.9	67.3 RH 87.0	71.7 RH 88.8	75.6 RH 89.2	77.9 RH 85.5	78.3 RH 84.7	75.7 RH 85.0	71.8 RH 84.1	65.8 RH 82.9	60.0 RH 83.1	
5 - 6 AM	57.0 RH 84.8	57.2 RH 85.7	60.4 RH 87.1	67.1 RH 87.0	71.7 RH 88.9	75.6 RH 89.0	78.0 RH 85.6	78.2 RH 84.9	75.7 RH 85.0	71.6 RH 84.2	65.8 RH 83.1	59.8 RH 83.2	
6 - 7 AM	56.9 RH 84.8	58.1 RH 85.6	60.6 RH 86.7	67.8 RH 86.0	72.5 RH 87.6	76.6 RH 87.3	79.0 RH 78.2	79.1 RH 83.2	76.5 RH 84.0	72.1 RH 83.6	65.9 RH 83.1	59.8 RH 83.2	
7 - 8 AM	57.7 RH 83.7	60.0 RH 84.8	61.9 RH 84.5	68.4 RH 82.0	74.3 RH 83.9	78.8 RH 82.3	81.7 RH 70.2	81.6 RH 78.3	77.3 RH 79.0	74.3 RH 79.2	67.1 RH 80.7	60.9 RH 81.5	
8 - 9 AM	59.5 RH 79.9	62.1 RH 80.5	64.0 RH 79.6	72.0 RH 77.0	76.7 RH 78.8	82.0 RH 75.0	85.3 RH 64.6	85.0 RH 71.3	82.2 RH 72.0	76.8 RH 73.6	68.9 RH 76.5	62.8 RH 77.0	
9 - 10 AM	61.5 RH 75.2	63.7 RH 76.1	66.0 RH 75.2	74.3 RH 72.0	78.5 RH 75.4	84.4 RH 70.0	88.0 RH 61.5	87.4 RH 66.5	84.6 RH 67.0	78.6 RH 69.9	70.5 RH 73.3	64.8 RH 73.0	
10 - 11 AM	63.2 RH 71.7	64.8 RH 72.1	67.8 RH 71.7	75.9 RH 69.0	79.6 RH 73.5	85.8 RH 67.7	89.7 RH 61.6	88.9 RH 63.6	86.0 RH 64.0	79.9 RH 66.7	71.4 RH 71.1	66.2 RH 71.1	
11 - 12 AM	64.5 RH 69.1	65.2 RH 70.1	68.9 RH 69.7	76.8 RH 67.0	80.3 RH 72.2	86.3 RH 67.4	90.5 RH 61.3	89.5 RH 63.0	86.7 RH 63.0	80.4 RH 65.7	72.0 RH 69.8	67.1 RH 65.7	
12 - 1 PM	64.7 RH 68.5	65.0 RH 69.2	69.3 RH 69.3	77.2 RH 65.0	79.5 RH 72.7	86.5 RH 68.0	90.6 RH 61.5	89.7 RH 62.9	86.8 RH 64.0	80.6 RH 65.7	71.9 RH 69.7	67.3 RH 67.4	
1 - 2 PM	64.4 RH 69.0	64.3 RH 69.1	68.9 RH 69.6	76.6 RH 68.0	79.7 RH 73.3	85.9 RH 69.0	90.3 RH 62.3	89.3 RH 64.1	86.4 RH 64.0	80.1 RH 66.6	71.5 RH 69.6	67.0 RH 67.3	
2 - 3 PM	63.7 RH 70.3	63.3 RH 70.6	67.8 RH 71.1	75.9 RH 69.0	79.1 RH 74.4	84.9 RH 71.4	89.4 RH 64.0	88.4 RH 65.7	85.5 RH 65.0	79.2 RH 67.5	70.7 RH 71.5	66.2 RH 68.7	
3 - 4 PM	62.8 RH 72.3	62.2 RH 71.9	67.1 RH 72.8	74.8 RH 71.0	77.6 RH 75.8	83.6 RH 74.1	88.3 RH 65.7	87.4 RH 67.7	84.3 RH 67.0	77.9 RH 69.7	69.8 RH 73.5	65.1 RH 70.9	
4 - 5 PM	61.7 RH 74.4	61.2 RH 75.0	66.2 RH 74.6	73.8 RH 73.0	77.4 RH 77.3	82.6 RH 75.6	87.2 RH 68.0	86.3 RH 69.6	82.7 RH 71.0	76.5 RH 72.6	68.9 RH 75.2	64.0 RH 73.2	
5 - 6 PM	60.7 RH 77.2	60.5 RH 77.5	65.1 RH 77.3	72.5 RH 74.0	76.4 RH 79.7	81.5 RH 77.7	85.4 RH 71.4	84.7 RH 72.5	81.3 RH 74.0	75.6 RH 73.1	68.1 RH 77.6	63.1 RH 76.1	
6 - 7 PM	60.2 RH 78.9	59.9 RH 79.3	64.2 RH 79.5	71.4 RH 78.0	75.4 RH 81.6	80.4 RH 79.9	84.1 RH 73.4	83.4 RH 75.5	80.2 RH 77.0	74.8 RH 75.0	67.7 RH 78.6	62.6 RH 77.5	
7 - 8 PM	59.6 RH 80.4	59.4 RH 80.7	63.7 RH 80.9	70.9 RH 80.0	74.8 RH 83.0	79.7 RH 81.8	83.2 RH 76.0	82.6 RH 77.2	79.6 RH 78.0	74.3 RH 78.5	67.4 RH 79.6	62.2 RH 78.2	
8 - 9 PM	59.2 RH 81.3	59.1 RH 82.1	61.2 RH 82.6	70.3 RH 81.0	74.2 RH 84.3	79.1 RH 83.1	82.6 RH 78.0	81.9 RH 78.7	79.0 RH 79.0	73.9 RH 77.0	67.2 RH 79.6	61.9 RH 79.2	
9 - 10 PM	58.9 RH 81.8	58.9 RH 83.4	62.8 RH 83.4	69.8 RH 83.0	73.8 RH 85.2	78.6 RH 84.1	81.9 RH 79.2	81.4 RH 79.7	78.5 RH 81.0	73.6 RH 80.3	67.0 RH 80.0	61.6 RH 80.1	
10 - 11 PM	58.6 RH 82.5	58.6 RH 83.5	62.4 RH 76.2	69.4 RH 84.0	73.5 RH 85.8	78.1 RH 85.1	81.2 RH 80.6	80.9 RH 80.6	78.1 RH 81.0	73.2 RH 81.1	66.8 RH 80.5	61.4 RH 80.5	
11 - 12 PM	58.3 RH 83.2	60.4 RH 83.9	63.1 RH 85.1	69.1 RH 85.0	73.1 RH 86.8	77.6 RH 86.3	80.5 RH 82.1	80.3 RH 81.9	77.7 RH 82.0	73.0 RH 81.9	66.6 RH 81.0	61.1 RH 81.3	

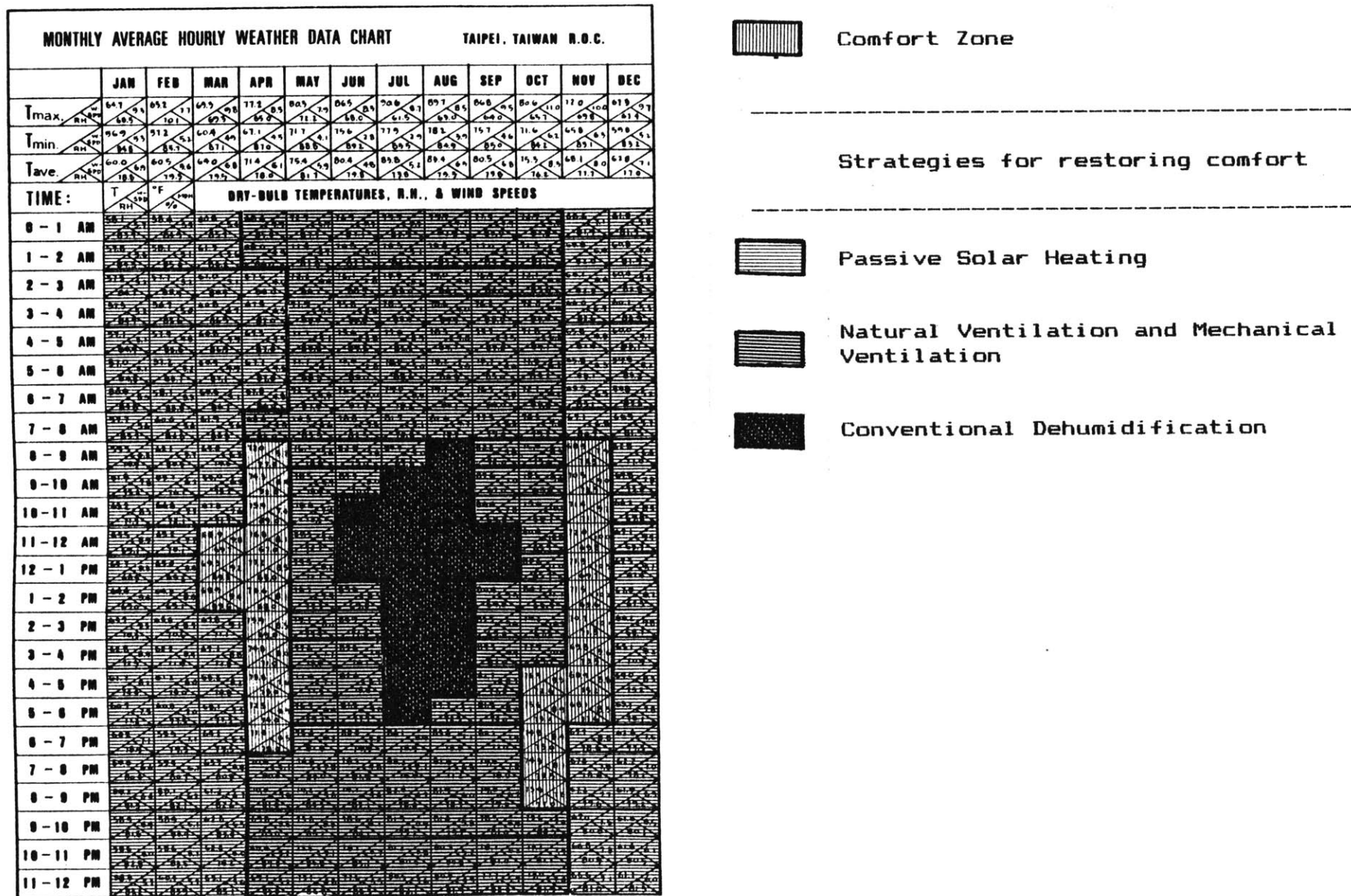
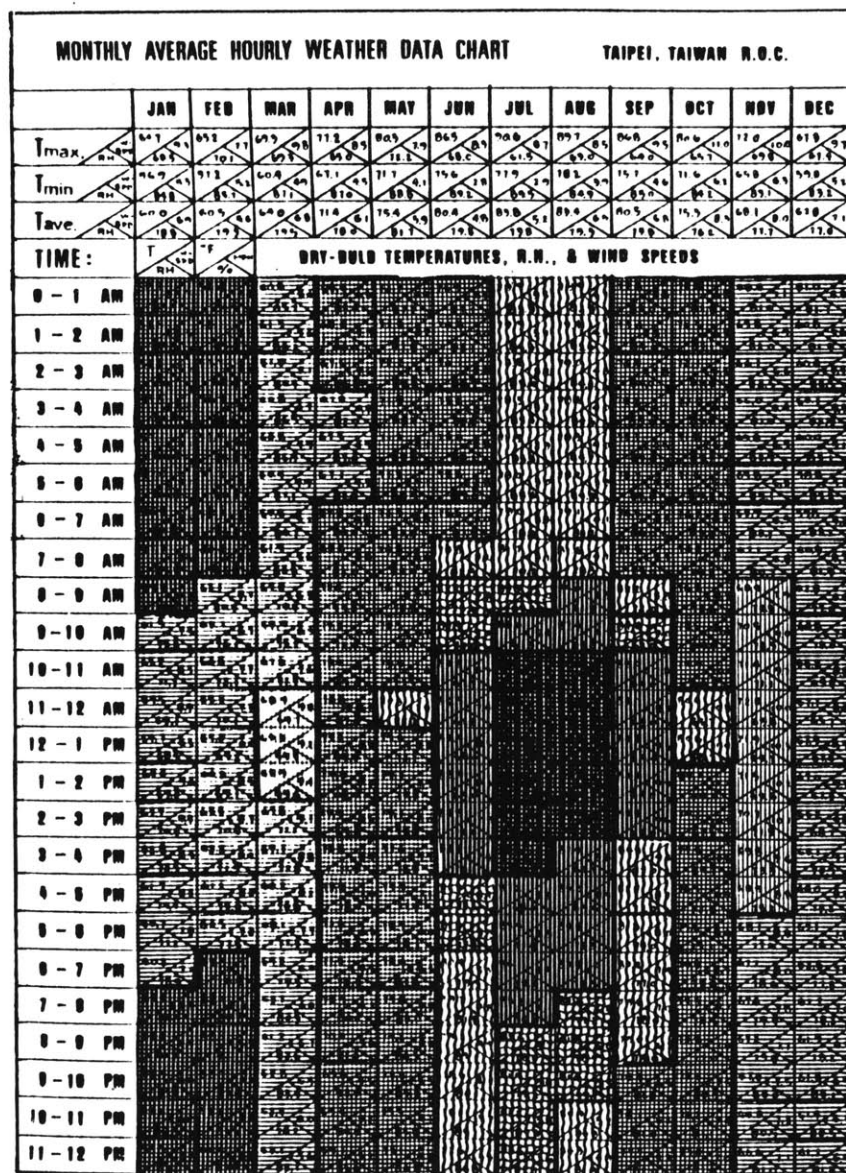


Fig. 3.8 Human Comfort Needs on the M.A.H.W.D. Chart as Analyzed by Givoni's Psychrometric Chart



Comfort Zone

Strategies for restoring comfort

Passive Solar Heating



50 - 100 Btuh



0 - 50 Btuh

Natural Ventilation and Mechanical Ventilation



0 - 100 ft/min (wind speeds)



100 - 200 ft/min



200 - 300 ft/min



300 - 500 ft/min



Conventional Dehumidification

Fig. 3.9 Human Comfort Needs on the M.A.H.W.D. Chart as Analyzed by Aren's Bioclimatic Chart

The sky-dome sun chart (Fig. 3.11) is a horizontal projection of the sunpath as seen from the apex. The altitude angles are shown at the 10 degree intervals by concentric circles. They range from 0° at the outside circle (Horizon) to 90° at the center point (Apex). The azimuth angles are shown at 10 degree intervals by equally spaced radii; They range from 0° (or 360°) at north meridian to 180° at the south meridian. The observation point is in the center of the chart (Apex). The elliptical curved lines in the chart represent horizontal projections of the sunpaths.

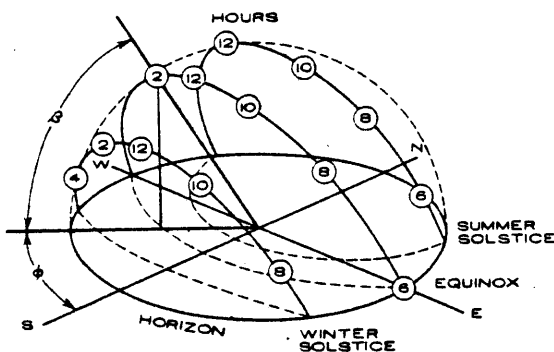


Fig. 3.10 Solar Altitude and Azimuth

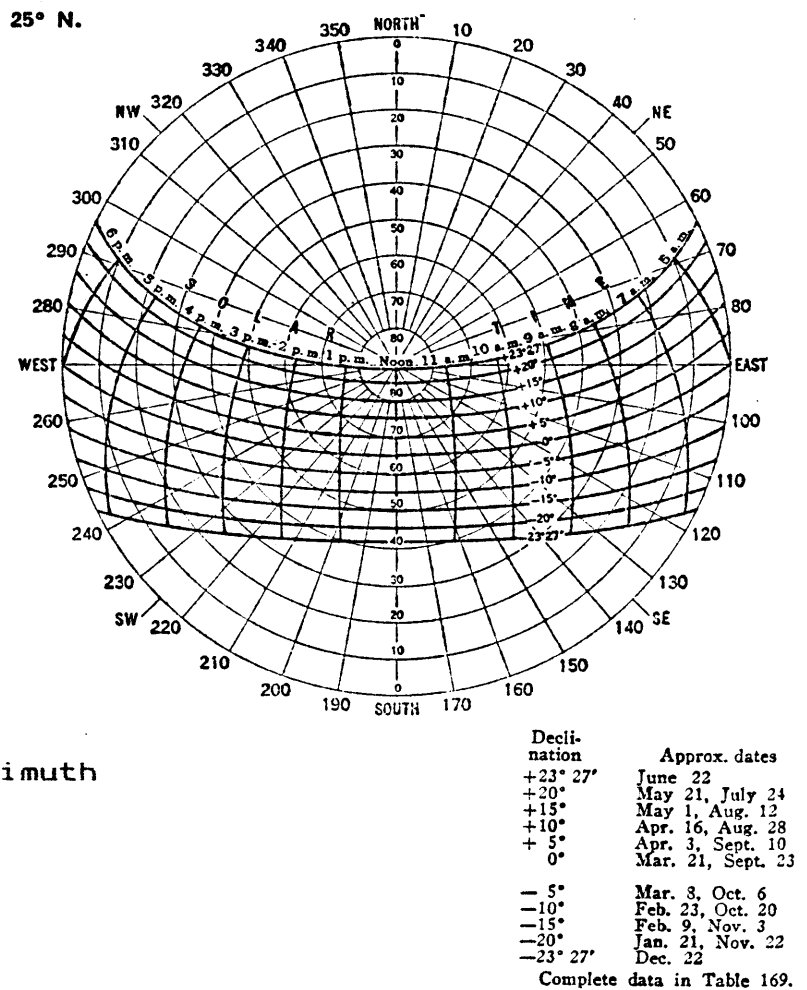


Fig. 3.11 Sky-Dome Sun Chart

The cylindrical sun chart (Fig. 3.12) is a vertical projection of the sunpath as seen from the earth. The chart was constructed with 0° latitude as the horizontal axis and 180° azimuth (true south) as the vertical axis. The observation point is the intersection of these two axes. The altitude angles are shown at 10 degree intervals by lines which are parallel to the horizontal axis. They range from 0 at the bottom (horizon) to the 90° at the top. The azimuth angles are shown at 10 degree intervals by lines which are parallel to the vertical axis. They range from 180° at the center to 0° or 360° at both sides. The sunpath along the sky vault is plotted on the chart as elliptical curved lines.

Two points must be kept in mind when using the sun chart. First, the azimuth angles shown on the chart are measured from true or geographic north which differs, by a constant angle depending on longitude, from magnetic north as read on a compass. Secondly, the times referred to on the chart are solar times, which may differ from the local standard time. When Daylight Savings Time is in effect, one hour should be subtracted from local time.

2.4.2 Shading Mask Protractor ,

The effect of shading devices can be plotted in the same manner as the sunpath was projected. The diagram that shows which part of the sky vault will be obstructed by the shading devices are called "shading masks". Shading masks

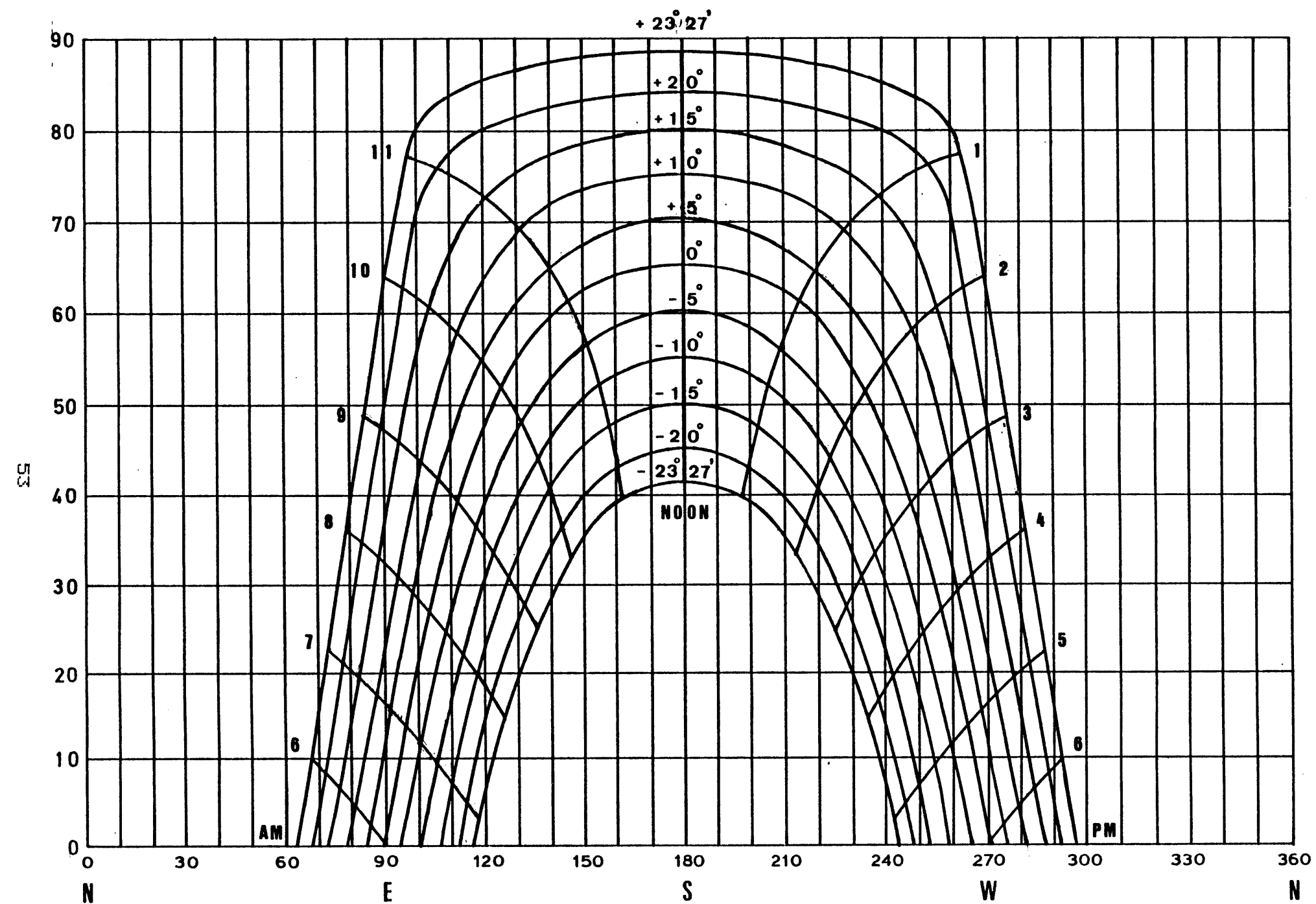


Fig. 3.12 Cylindrical Sun Chart

will change at different observation points on the window (Fig. 3.13), they are geometric projections and as such, are independent of latitude and exposed orientations. As a result, they can be used with equal accuracy in any direction and at any location.

The shading mask protractor is used to design shading devices for the periods of overheating. The protractor is in the same projection and scale as the sun chart, therefore, it is useful to overlay a transparent copy of the protractor onto the local sun chart to predict shading effect of a particular overhang or vertical fin design.

Fig. 3.14 shows a shading mask protractor for the sky-dome sun charts. The upper part of the sun chart showing bearing angles is used to determine the shading masks for vertical fins. The lower part showing curved lines is titled "profile angles", and is used to determine the shading masks for horizontal overhangs.

Fig. 3.15 shows the shading mask protractor for cylindrical sun charts. The curved lines are used to determine the shading masks of horizontal overhangs. Imaginary vertical lines can be used to determine the shading masks of vertical fins. Fig. 3.16 shows two different types of shading masks for a given shading device.

A set of shading devices and their shading masks illustrated by Olgyay in 1963 is widely accepted - Architecture Graphic Standards used them as an illustration of shading device design. These shading masks however,

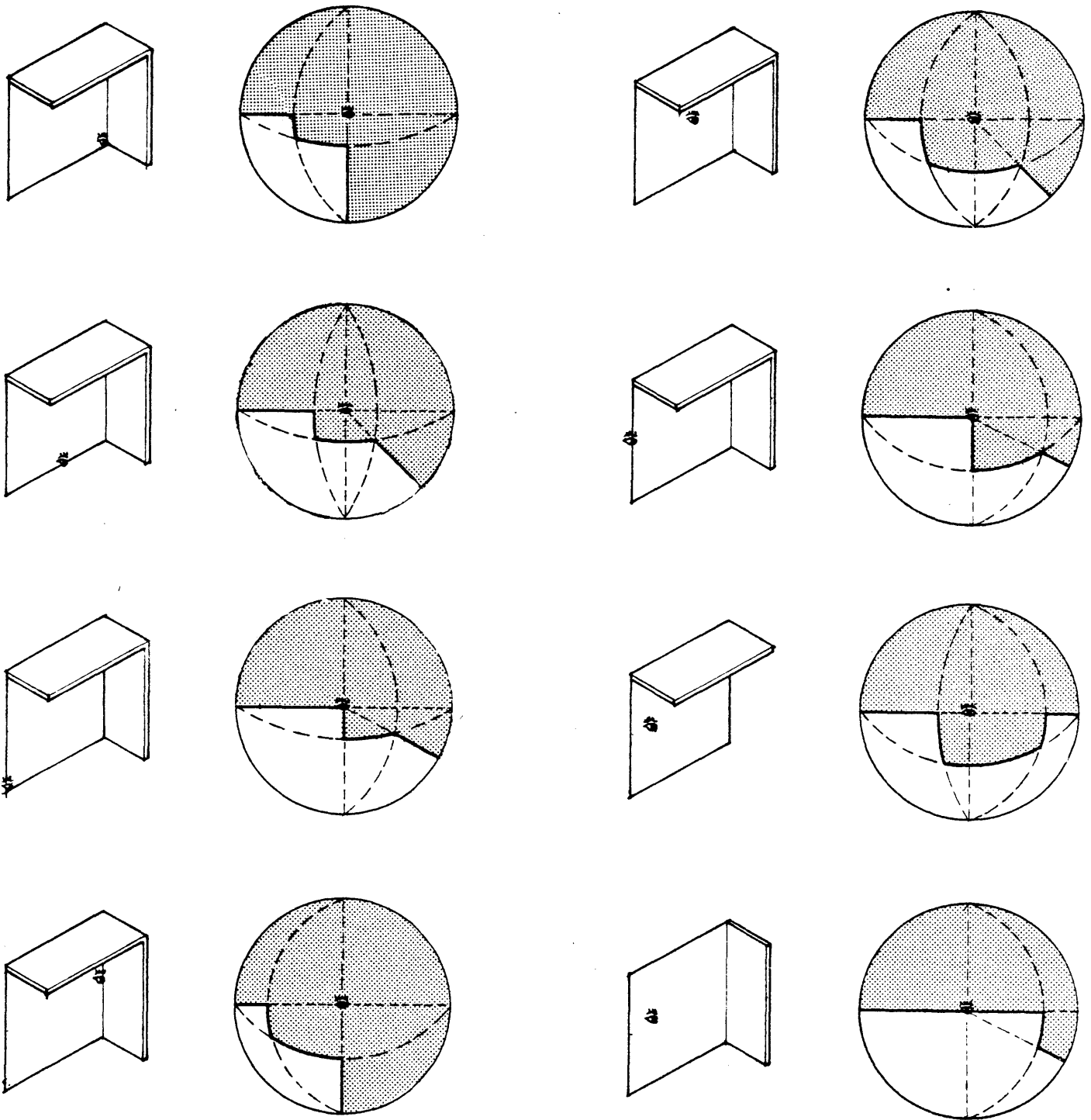


Fig. 3.13 Sky-Dome Shading Masks with Different Observation Points

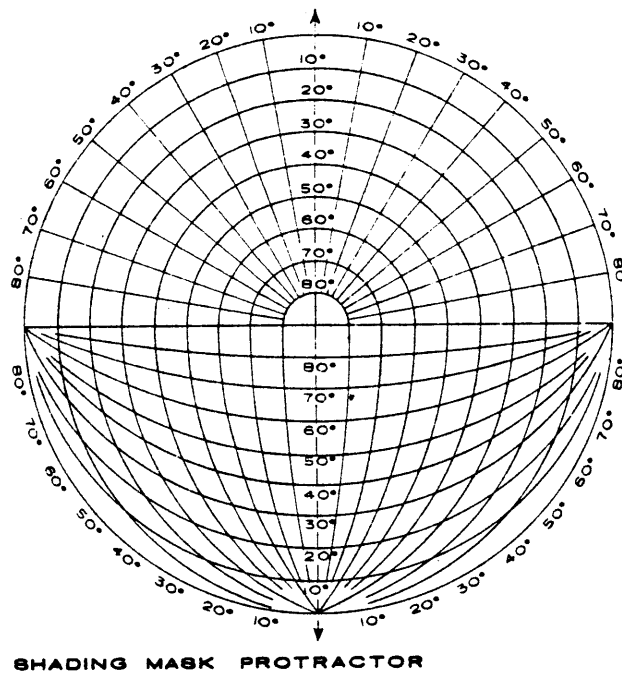


Fig. 3.14 Sky-Dome Shading Mask Protractor

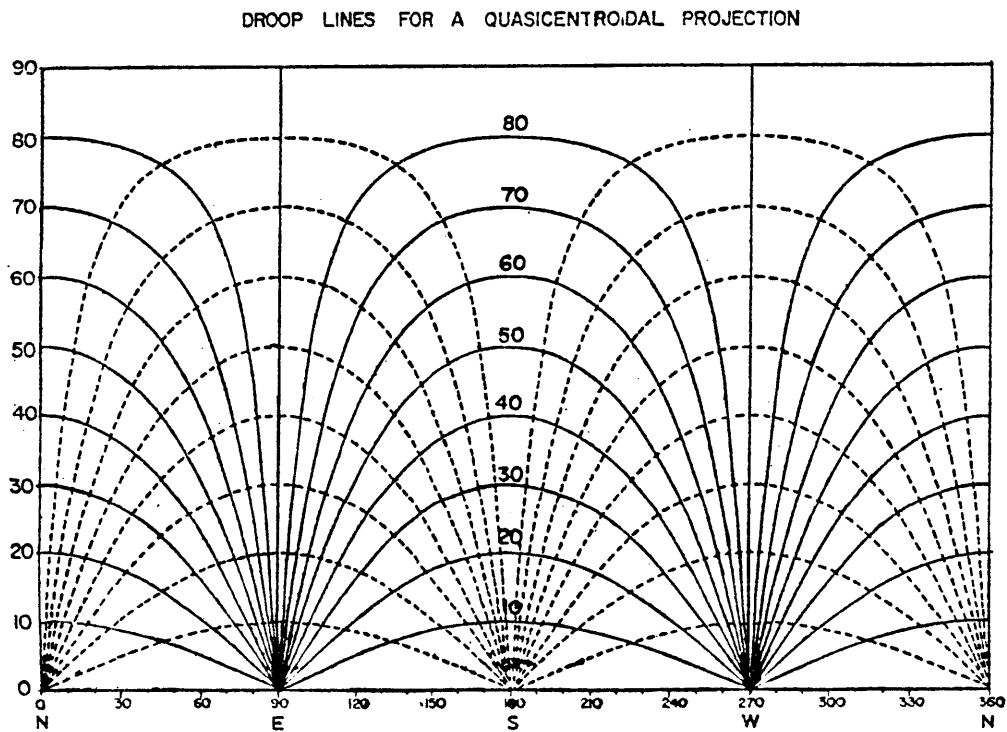


Fig. 3.15 Cylindrical Shading Mask Protractor

HORIZONTAL

VERTICAL

EGGCRATE

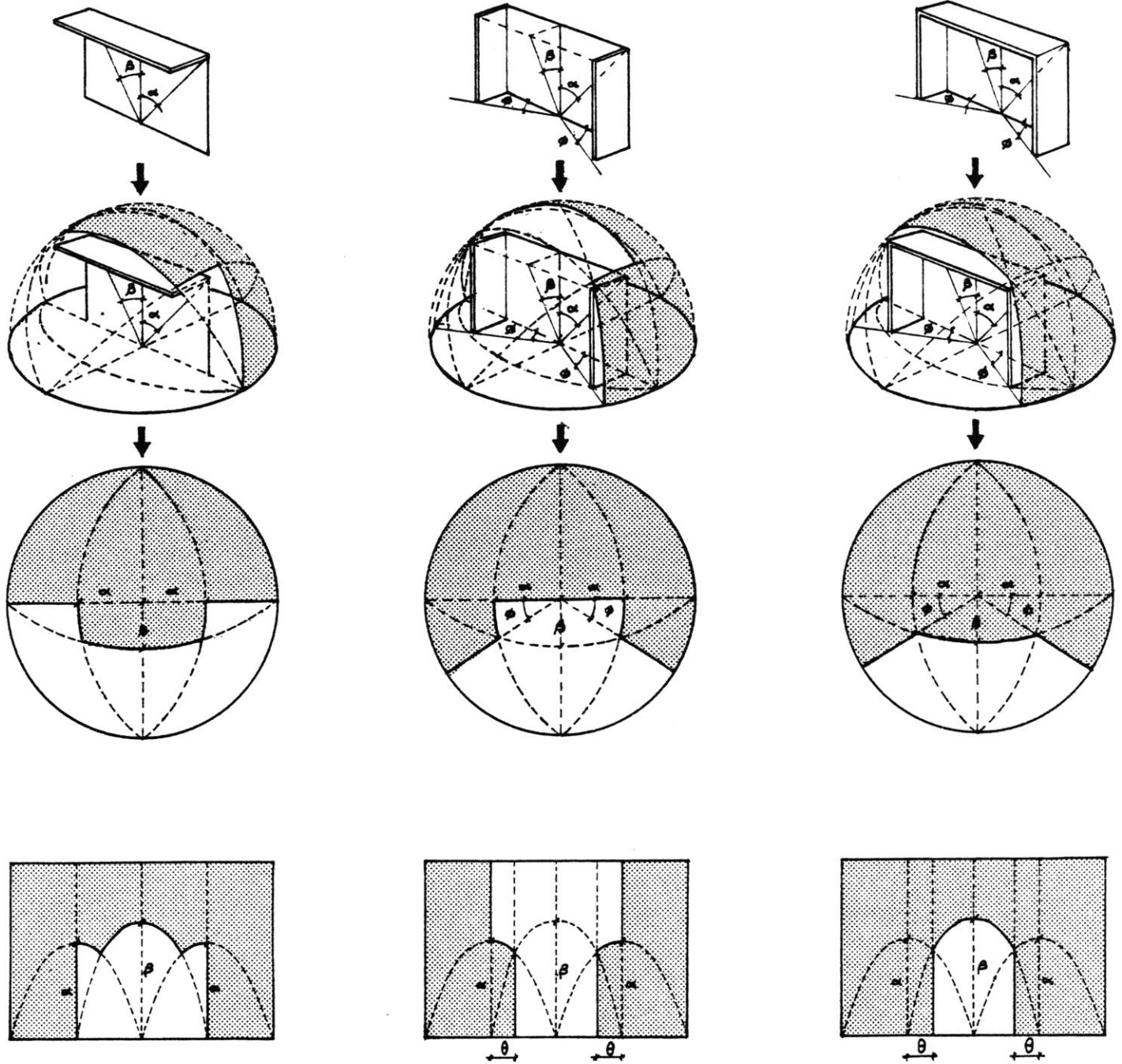


Fig. 3.16 Sky-Dome and Cyclindrical Shading Masks for the Same Shading Devices

except for the egg crate type of shading device, are not completely correct. They are wrong because the horizontal overhangs or the vertical fins never have infinite length. Corrections must be made to represent the real shading masks determined by their relative shading device (Fig. 3.17).

2.4.3 "Shaded" Sun Chart and Shading Devices Design

Overheated periods can first be analyzed by plotting M.A.H.W.D. on the psychrometric or bioclimatic chart, and then recorded on a M.A.H.W.D. chart as mentioned before. These periods can also be superimposed on a sun chart to determine the needed shading masks.

Often, months in which the sun follows the same path will have different shading needs. For example, September 23 and March 21 have the same sunpath but shading is needed all day long in September, but only from 1am to 3pm in March. Generally speaking, the fall months are warmer and need shading while the spring months are colder and need sunshine. This is due to the thermal mass of the earth which absorbs heat in summer or becomes cold in winter and thereby influences the ambient temperatures in fall or spring. When shading needs are different but the sunpaths are similar during these months, seasonally adjusted shading devices are needed.

Fig. 3.18 shows the shading needs for Taipei, the heavily dotted area shows permanent shading needs. The lightly dotted area shows seasonally adjusted shading needs.

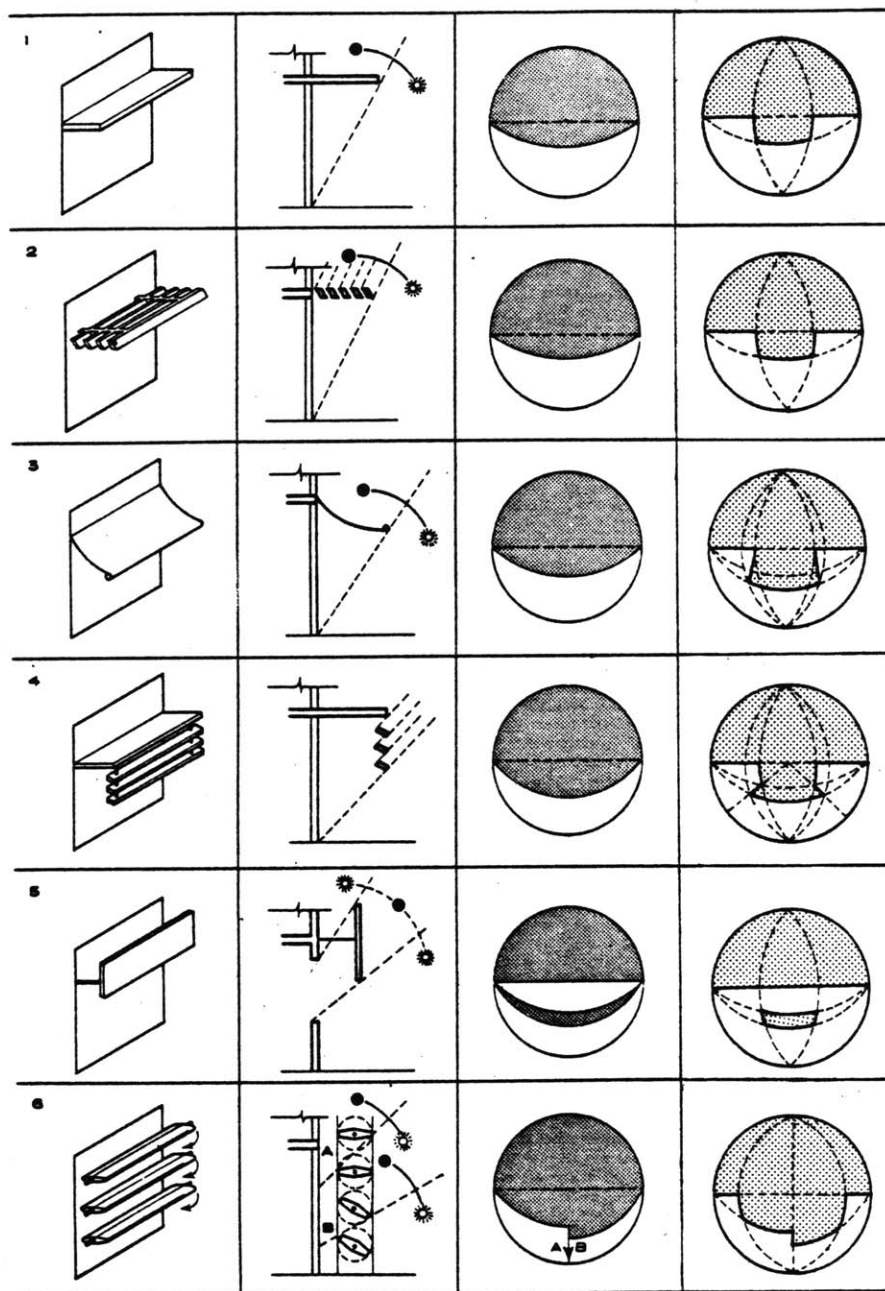


Fig. 3.17 Corrections to Olgyay's Shading Masks

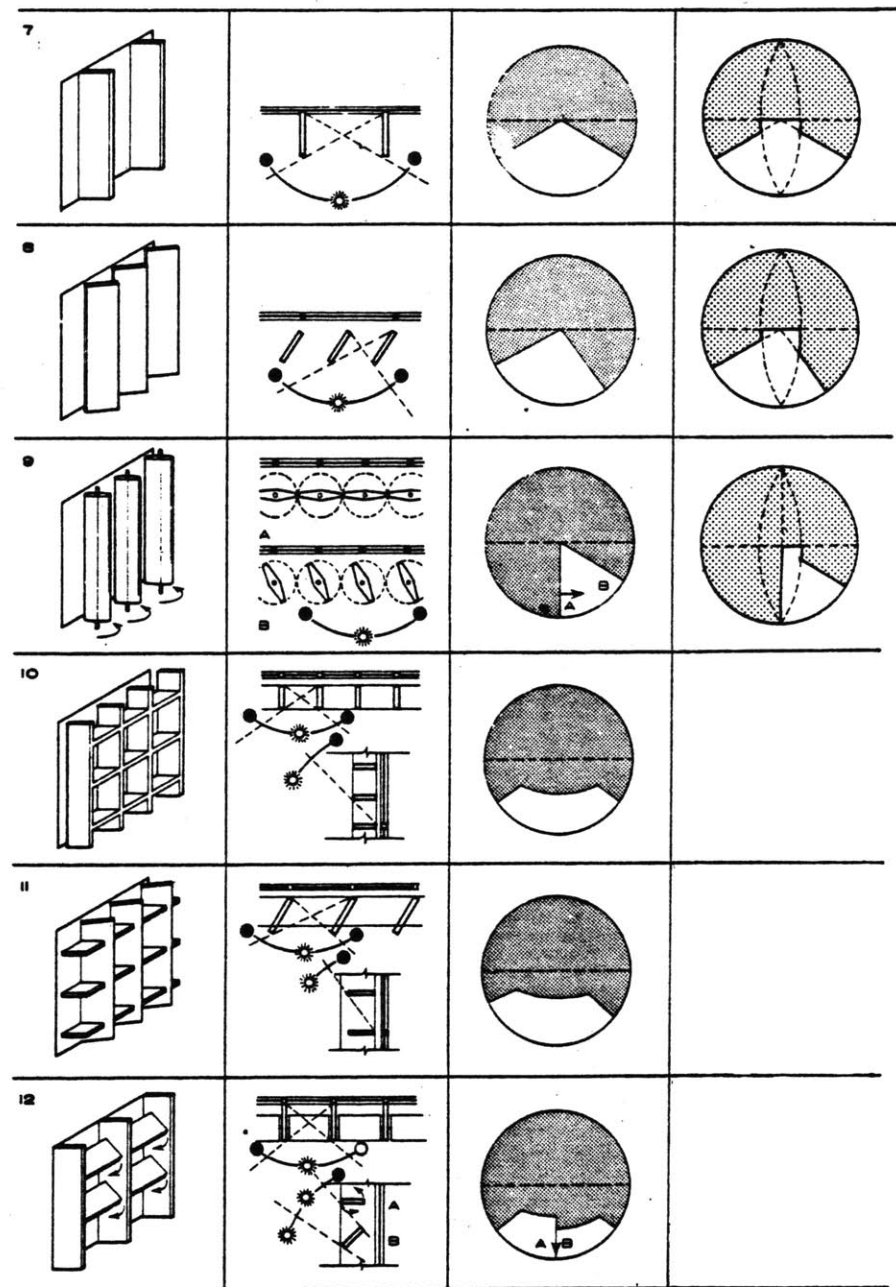


Fig. 3.17 Corrections to Olgyay's Shading Masks

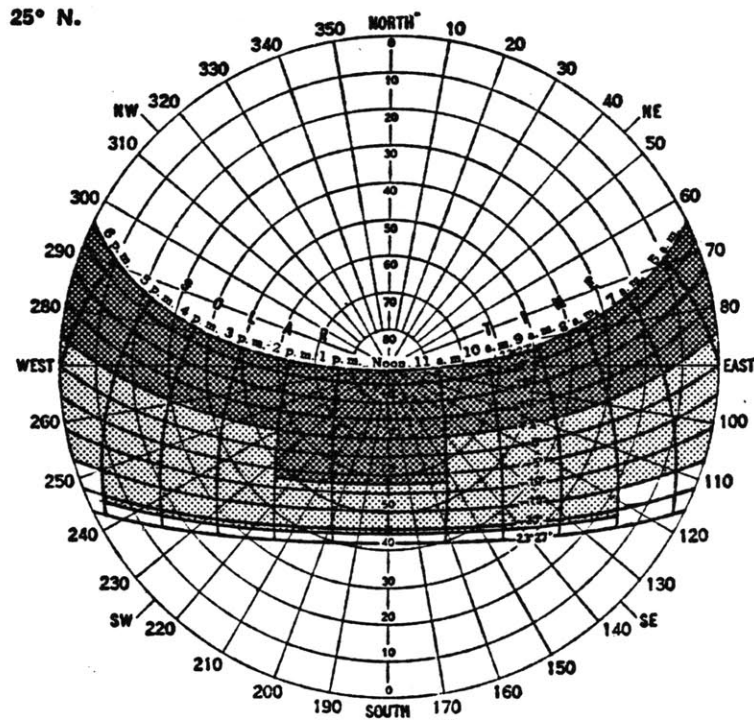


Fig. 3.18 "Shaded" Sun Chart for Taipei

If the orientation of a window is known, this "shaded" sun chart and shading mask protractor can be used to design climatically responsive shading devices for the window. A set of examples of this method will be presented in Chapter IV.

2.5 Wind Roses

Wind roses for a specific area show the frequency in percent, of winds from different compass directions. They are most helpful in passive cooling designs where natural or wind induced ventilation is appropriate. In such cases,

architects can use wind roses to determine the optimal building orientation and in designing the "wind inducing" devices.

In general, the desired summer winds should be utilized for cooling, while the severe winter winds should be deflected. Fig. 3.19 shows the wind roses for Taipei, based on the weather data for the period 1972-1981.

2.6 Soil Temperature Profile Analysis

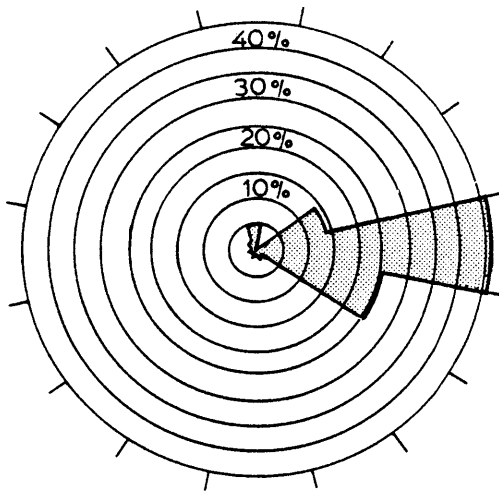
In order to determine the applicability of direct earth coupled cooling or detached earth coupled cooling strategies, ground temperature profiles at different depths are essential. Fig. 3.20 shows the ground temperature profiles of Taipei at 1, 12, and 25 ft. in comparison to the monthly maximum and minimum air temperature. These ground temperatures are calculated in Appendix C.

Because both the earth coupled cooling and the detached earth coupled cooling systems require that the interior surface temperatures at least 8 F lower than the desired room air temperature (78 F) in summer in order to cool, it can be seen that these strategies do not apply to Taipei.

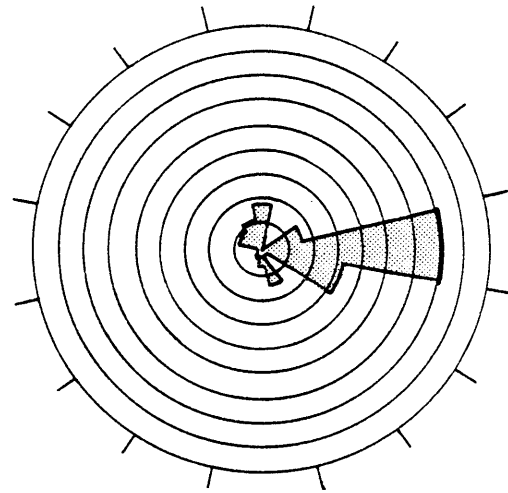
2.7 Cooling Load Calculations

Summer cooling loads in hot, humid climates can be grouped into two categories: sensible loads and latent loads. Sensible cooling load is the rate at which heat must be removed from the space to maintain room air temperature

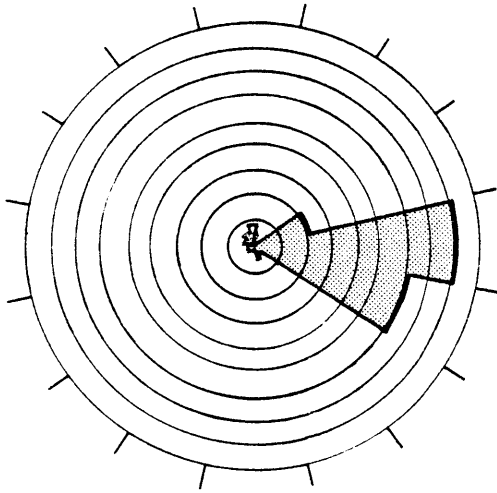
JAN



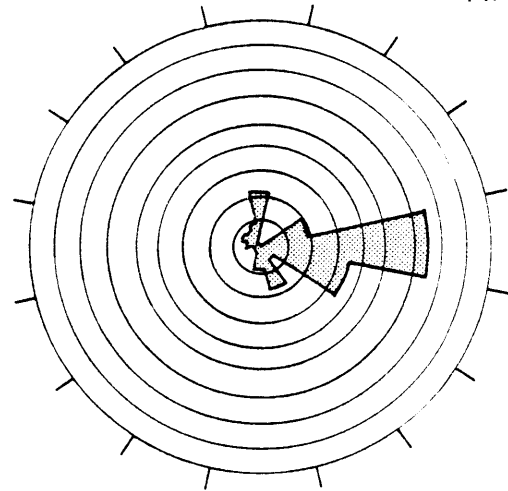
APR



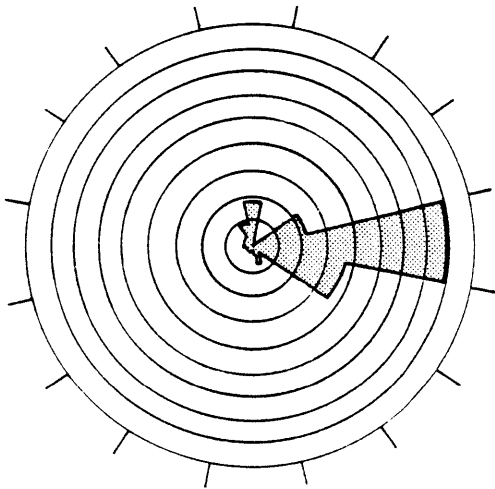
FEB



MAY



MAR



JUN

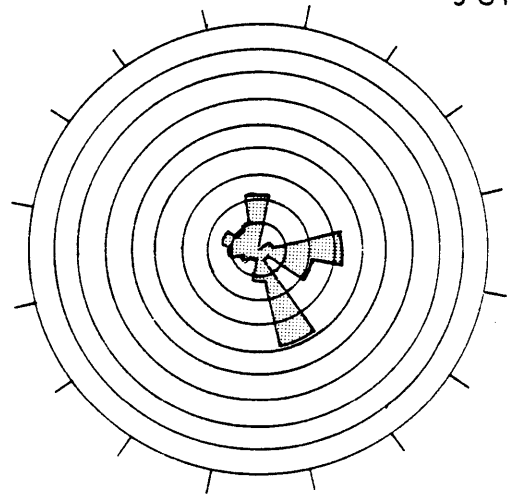


Fig. 3.19 Wind Roses of Taipei, for the Time Period:
1972-81

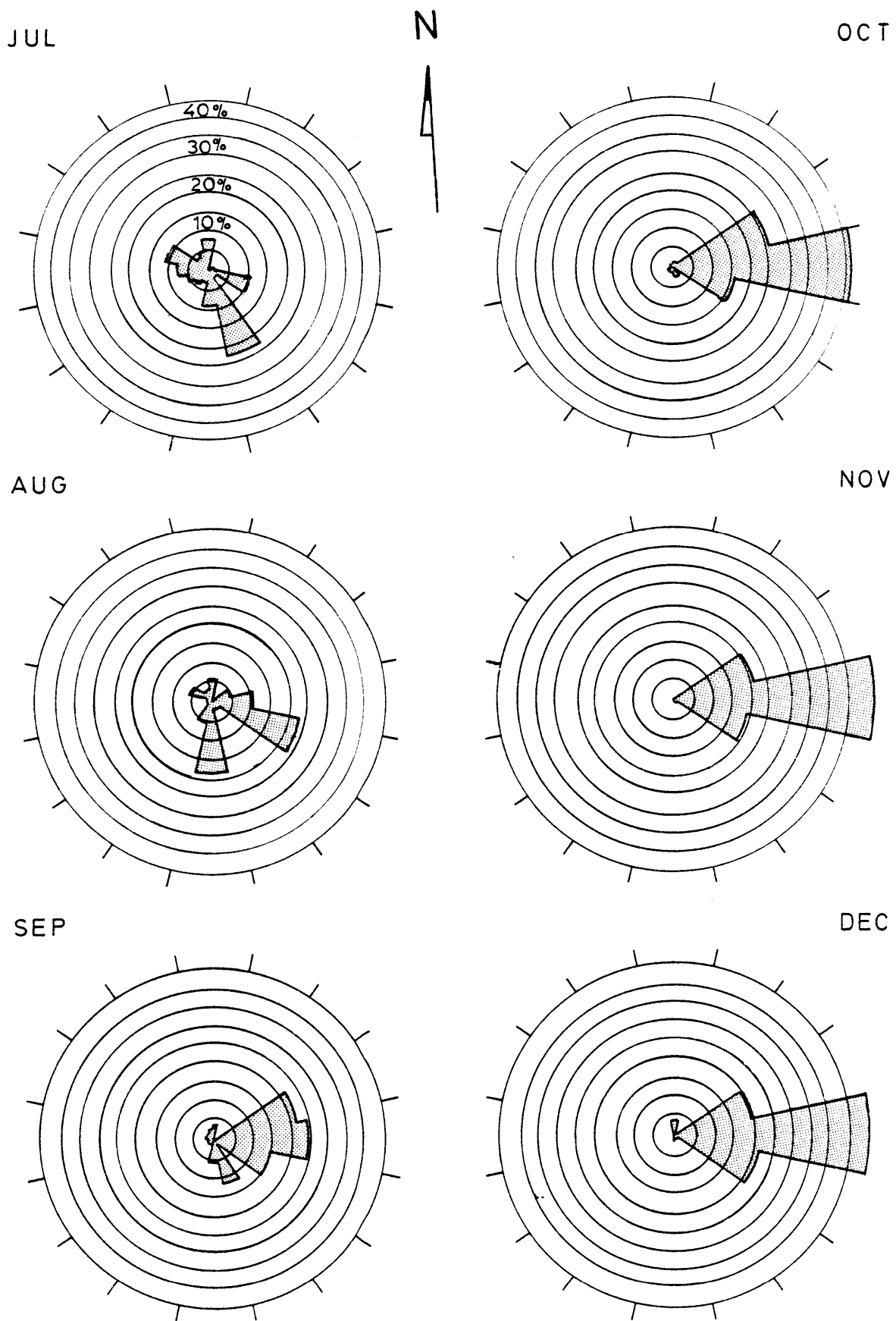


Fig. 3.19 Wind Roses of Taipei, for the Time Period:
1972-81

at a constant value (usually 78°F for residences). Latent cooling load is the rate at which moisture of the air must be removed from the space to maintain room air absolute humidity at constant value (usually 72 grains in one pound of dry air, i.e., 50% R.H. at 78°F). These cooling loads are not only affected by the external heat gains determined by the weather conditions and envelop design (load control strategies), but they are also influenced by internal heat gains such as the heat released from occupants, lights, or mechanical equipments, etc. (see Fig. 3.21) Table 3.3 shows the major cooling load equations developed by 1981 ASHRAE Fundamentals.

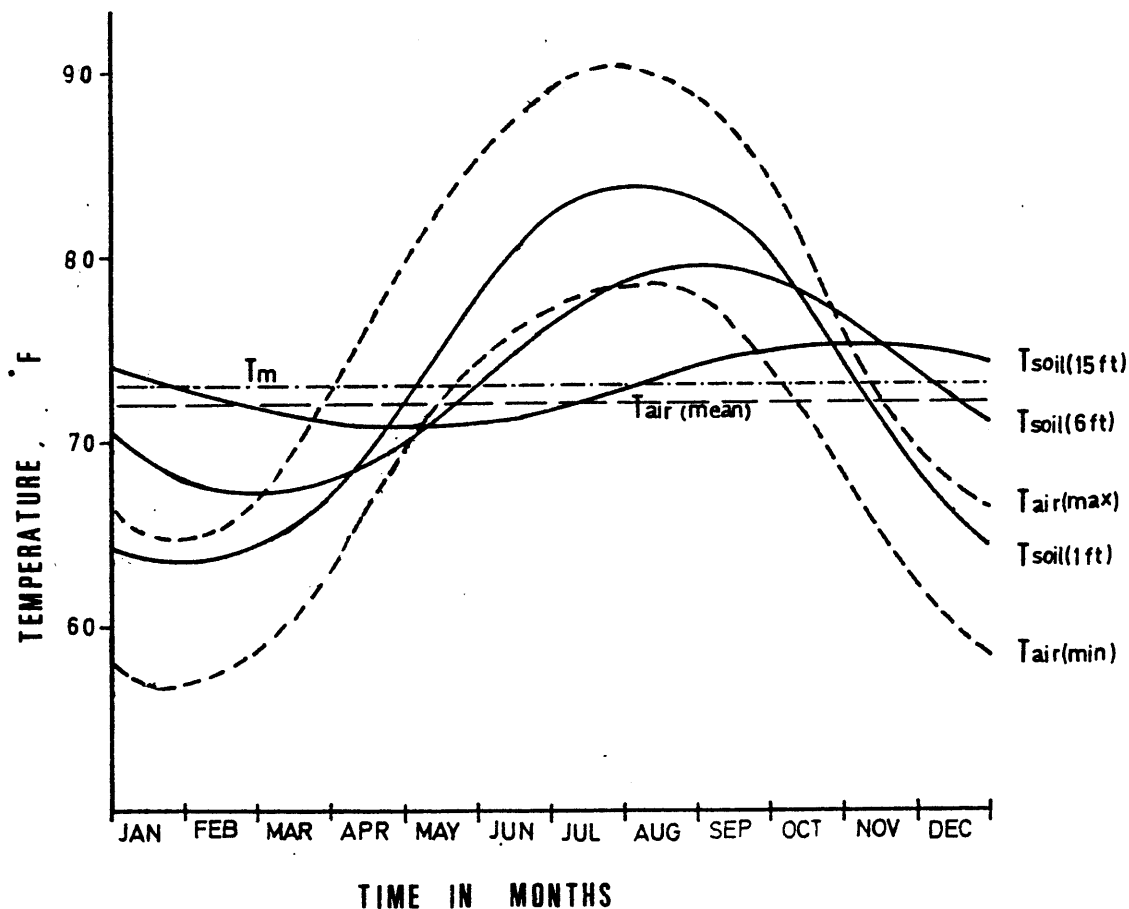


Fig. 3.20 Ground Temperature Profiles for Taipei

BUILDING HEAT LOSS/GAIN—HEAT GAIN IN SUMMER

The major heat gain factors in summer are depicted below

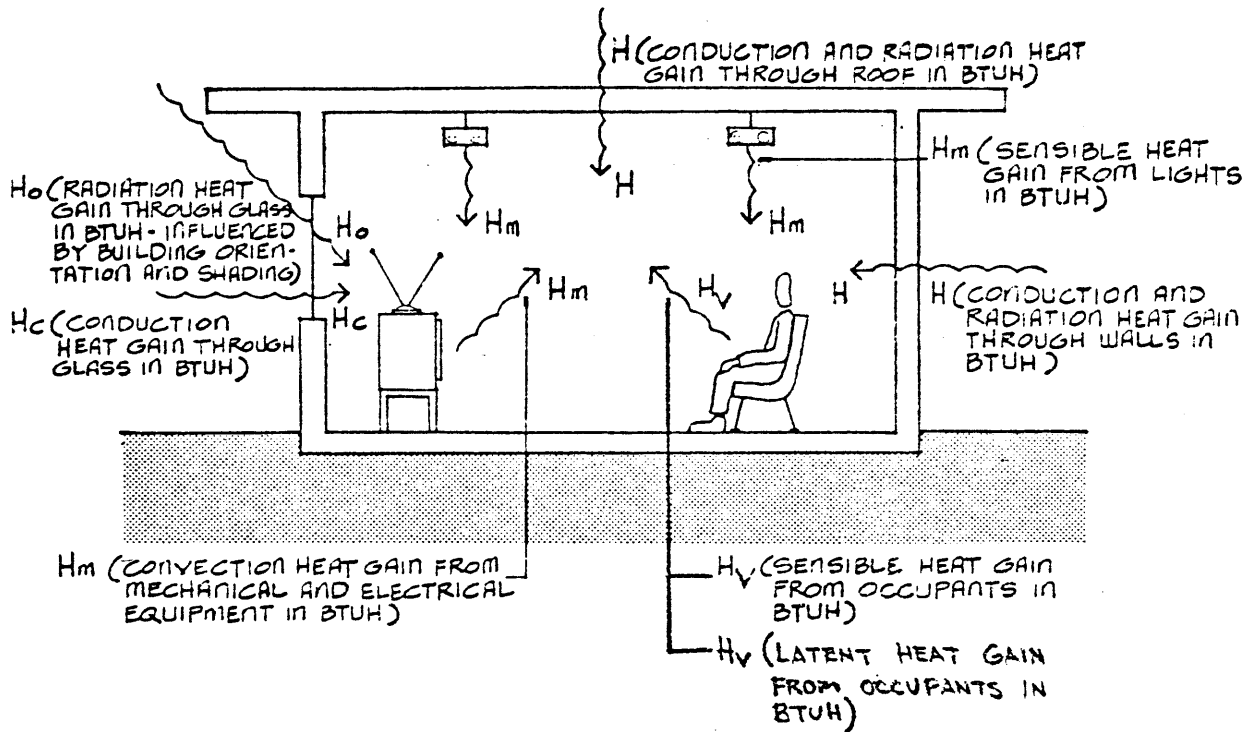


Fig. 3.21 Building Heat Gains in Summer

Basic Cooling Load Equations	Description
1) $Q = U * A * CLTD$	<p>U: Overall coefficient of heat transmission in Btuh / sqft-F</p> <p>A: Area of wall, roof, or glass surface in sqft</p> <p>CLTD: Cooling load temperature difference in F (Tair - Tsol-air)</p>
2) $Q = A * S.C. * SHGF$	<p>A: Area of openings in sqft</p> <p>S.C.: Shading coefficient (no units)</p> <p>SHGF: Solar heat gain factor in Btuh/sqft</p>
3) $Q = 3.4 * W$	W: Lighting, electrical, and mechanical equipment total energy in watts
4) Infiltration	
$Q_{sensible}$ $= 1.10 * CFM * T$	<p>CFM: Ventilation and Infiltration air, cubic feet per minute</p> <p>T: Inside-outside air temperature difference in °F</p>
Q_{latent} $= 1.10 * CFM * T$	<p>W: Inside-outside air humidity ratio difference, lb H2O/lb dry air</p>
5) Occupants	
$Q_{sensible} = 225 * (\text{number of occupants})$ $Q_{latent} = 225 * (\text{number of occupants})$	

Table 3.3

REFERENCES FOR CHAPTER III

1. ASHRAE Fundamentals, Published by the American Society of Heating, Refrigeration and Air Conditioning Engineers, 1977.
2. B., Givoni, "Man, Climate and Architecture", 2nd ed., (Applied Science Publishers, Ltd., London, 1976), chapter 16.
3. V., Olgyay, "design with Climate", (Princeton University Press, 1973), chapter 2.
4. E., Arens, "A New Bioclimatic Chart for Passive Solar Design", Proceedings of the American Section of the Internal Solar Energy Society, Volume 5.2, October 19-22, 1980, pp 1202-1206.
5. B., Novell, "A Simple Design Method for Shading Devices and Passive Cooling Strategies Based on Monthly Average Temperature", Passive Cooling, International Passive and Hybrid Cooling Conference, Miami Beach 1981, pp.392-396.
6. V., Olgyay, "Design with Climate", (Princeton University Press, 1973), chapter 7.
7. ASHRAE Fundamentals, Published by the American Society of Heating, Refrigeration and Air-Conditioning Engineers, 1977; chapter 25.

CHAPTER IV

DESIGN ISSUE AND PASSIVE SOLUTIONS

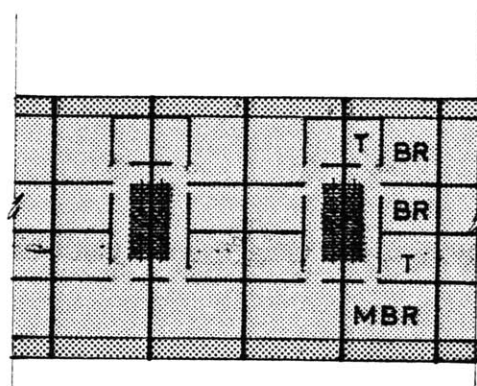
1.0 DESIGN ISSUE

The population density of population in third world countries is usually higher than that of the United States. Therefore, due to the limitation of space, multi-family housing units are very popular and quite necessary.

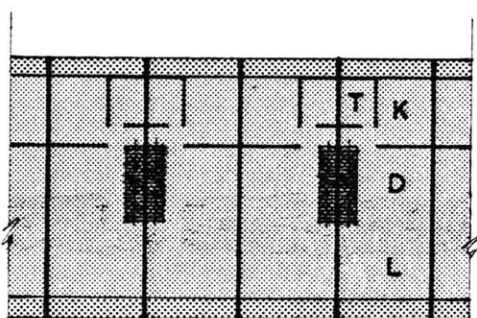
The five-story walk-up apartment is especially dominant in Taiwan. People usually use the first floor as a commercial shopping area with a sheltered walkway in front (by reason of high precipitation rates). The shopping area is divided by the building bays, and the owners always live in the apartment directly above. The prototype of this multifamily residence is depicted in Fig. 4.1.

These prototype housing designs, however, are usually established by manufacturers rather than by architects, as is shown by their similarity. Any rational design change in this type of housing reveals several difficulties with the original prototype.

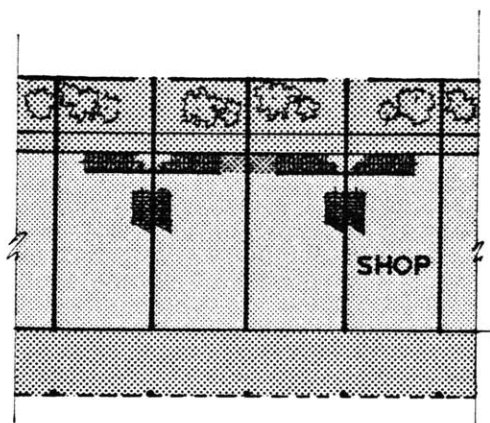
The popular prototype always uses the 4.5m - 6m bay as a division unit. Each unit has its own stairway leading up to the upper floors, and the housing unit above in this bay is always very uncomfortable on account of the narrow-bay arrangement which impedes natural ventilation. This "narrow-bay" type also produces another problem. People can



TYPICAL FL. PLAN



SECOND FL. PLAN



GROUND FL. PLAN

Fig. 4.1 The Prototype of Present Multifamily Residence in Taipei

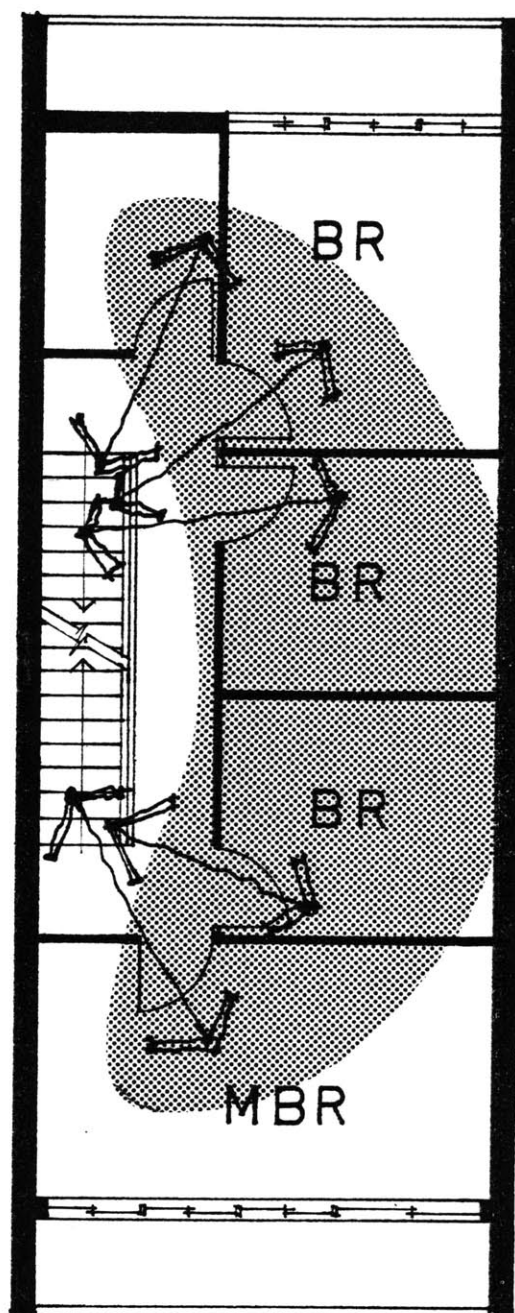


Fig. 4.2 Privacy Interruptions in Contemporary Building Designs

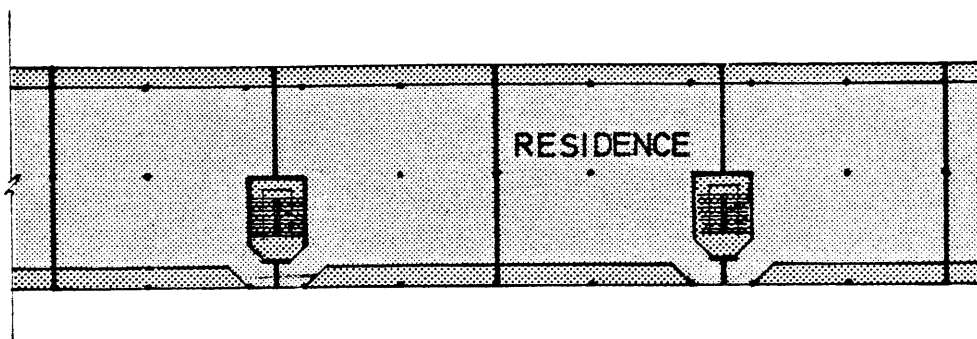
rarely afford to buy the 5 floors of a given bay unit, so, at least two families have to share each bay. This double occupancy causes great inconvenience because there is only one private stairway in each unit. Interruptions happen when the family has to go through the other family's private living space in order to use the stairway (see Fig. 4.2).

According to the scope of this thesis, there is no attempt made to change the overall design of this prototype but rather to suggest an acceptable design solution to existing problems. This solution would not only convince the manufacturer to accept very small changes in the architectural features (outward appearance, construction configuration, and 1 meter narrow but popular balcony, etc.), but would also greatly increase the feeling of comfort during the hot summer by allowing for passive cooling techniques. At the same time, this change would eliminate the present circulation or plan arrangement problems.

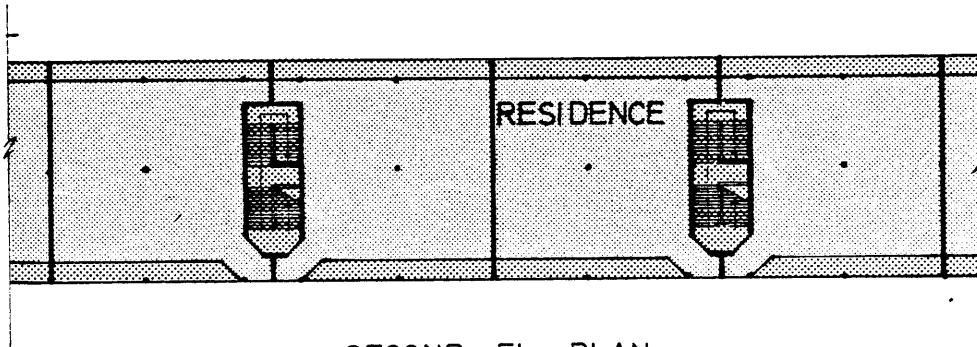
The following sections will introduce a series of design solutions which include climate responsive features allowing for the maximum benefit from proposed passive cooling techniques.

2.0 PLAN REARRANGEMENT

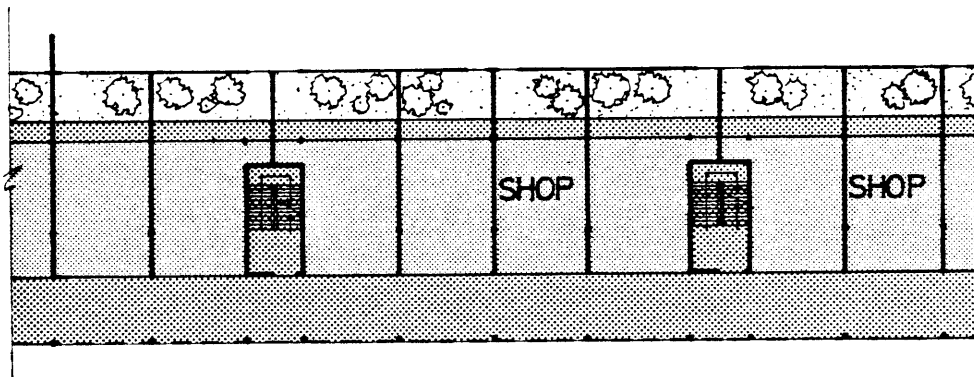
The newly designed plan (Fig. 4.3) does not change the outward appearance or the construction configuration (same



TYPICAL FL. PLAN



SECOND FL. PLAN



GROUND FL. PLAN

Fig. 4.3 A New Design Proposal

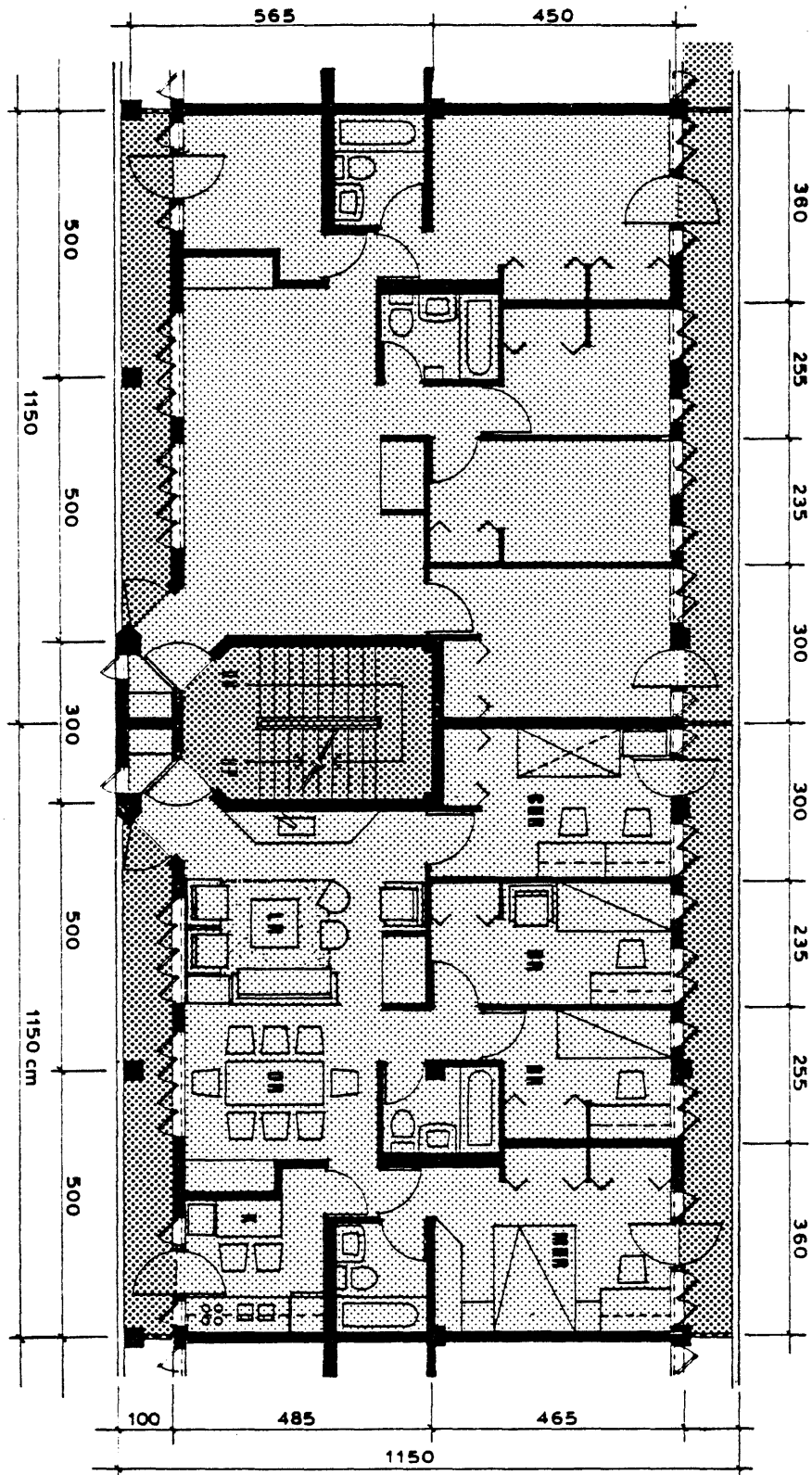


Fig. 4.3 A New Design Proposal

bay distance typ. 5 meter). Rather, than using individually distributed stairways, it uses stairways which are mutually shared by 8 families and 4 shops. The shopping area on the first floor retain the same popularly accepted pattern of street shops. The other floors of apartments, however, change dramatically from narrow, one bay units to units which are two bays wide. This new arrangement not only increases the natural cross-ventilation through this unit, but, also eliminates the waste of extra circulation area and its interruption of family privacy. The total unit area owned by a typical family remains the same as before. The new design simply removes the upper unit of each family, and puts the two units together on the same floor (see Fig. 4.4).

The example referred to in this thesis is a design for a 6-person extended family, as is the usual case in Chinese culture. The same principle of the proposed cooling techniques can also be applied to other multifamily housing types which accomodate different family sizes.

The rearranged plan, nearly without corridor space, is divided into two zones: living, dining, and kitchen areas are in one zone; bedrooms are in the other. This arrangement creates the greatest potential of using cross-ventilation to cool the occupants.

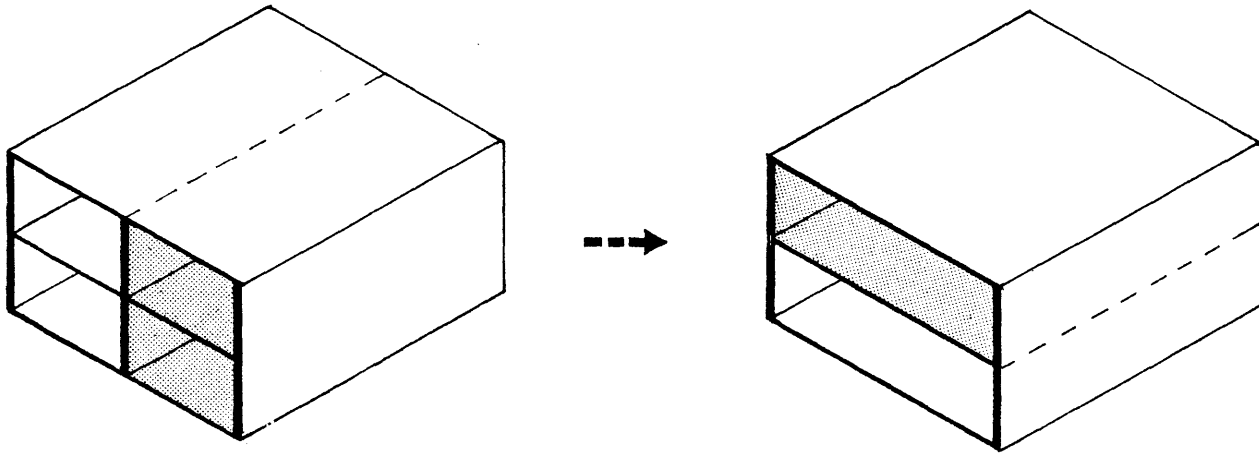


Fig. 4.4 A Design Concept for Preserving the Present Floor Area Per Family

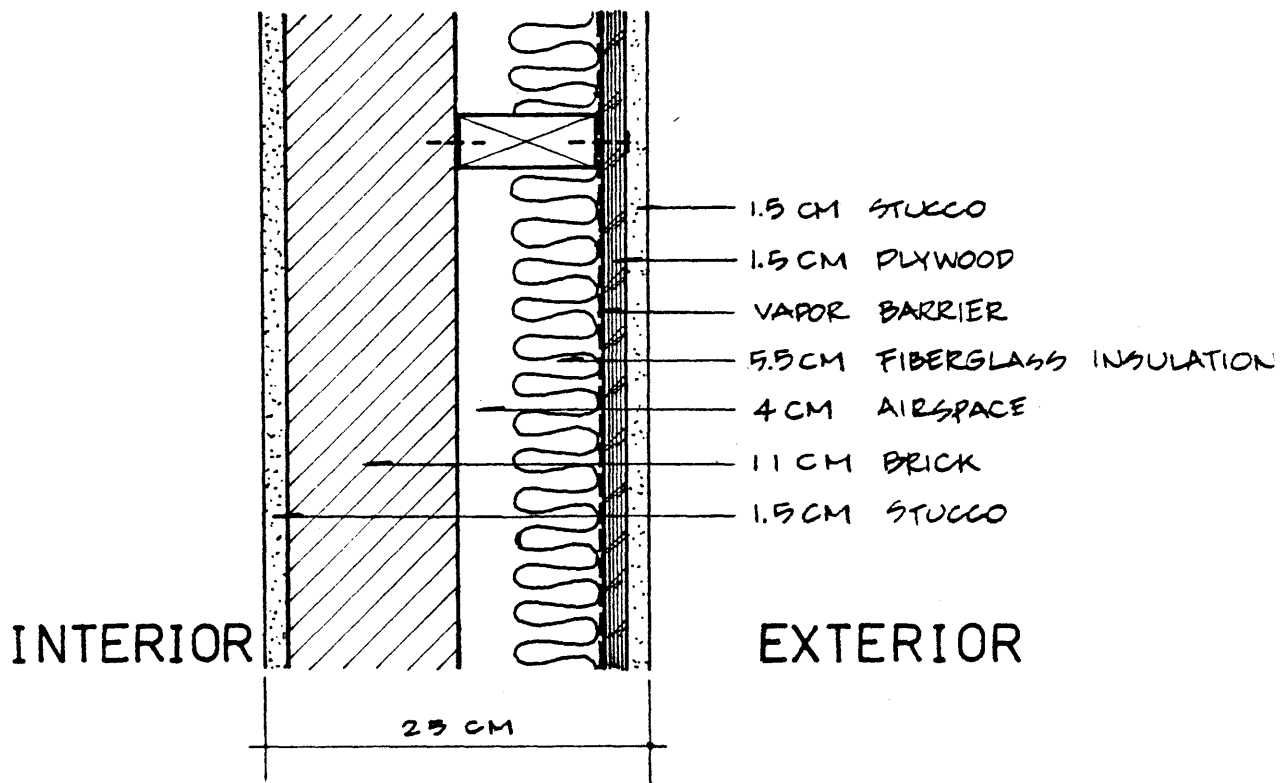


Fig. 4.5 An External Wall Section

3.0 LOAD CONTROL

3.1 Orientation

As mentioned in chapter II, it is best to control sun by orienting the building toward true south. It is difficult to encourage natural ventilation through the building, however, due to the prevailing wind direction which is from east during the spring or fall when natural ventilation is most essential for restoring comfort. Ventilation must be induced by special wind-inducing devices which will provide cross-ventilation.

In urban areas, a building's orientation is generally decided by the site and its relation to the street. Any orientation may happen in such a region as Taipei. Different sun control solutions and wind-inducing devices must therefore be individually designed for a specific site. These will be discussed in section 3.2 and 6.2 of this chapter.

3.2 Window Design

Window areas on both sides of the building are approximately the same to assure proper cross-ventilation (see Fig. 4.5). The total areas are minimized to reduce radiation and conduction heat gains while at the same time assuring good ventilation and daylighting. External louvered blinds are applied over the windows in the bedroom. They are also put on the lower parts of the windows in the living, dining and kitchen rooms to further reduce radiation

heat gains. The living room, dining room and kitchen are unobstructed so that the diffuse component of sunlight can be brought in for daylighting.

3.3 Sunshading Design

As discussed in Chapter III, section 2.4, with sun chart and a determination of the overheated periods we can know which parts of the year the windows should be totally shaded. And by this method we know which part of the sky vault should not be seen from any point on the window. Once this "shaded" sun chart is overlain by a shading mask protractor facing different directions we can design the optimal shading devices.

One point should be noted, that windows in this type of building facing any direction already have horizontal overhangs with approximately a 60° profile angle, because the prototype of the unit has a traditional 1 meter balcony in front of the window. This balcony is the overhang for the windows on the floor below. Eight differently orientated shading devices for north-south, east-west, NE-SW, and NW-SE oriented buildings are shown respectively in Figs. 4.8, 4.9, 4.12, & 4.13

4.0 CHOICE OF MATERIALS AND COLORS

The amount of insulation in the walls, roof (top floor), and doors must be calculated to reduce the conduction cooling load. Wall insulation should be placed as

close to the outside surface as possible rather than against the inside surface as is common practice in cold climates. This configuration is best in hot, humid climates, because the insulation so placed can eliminate the accumulation of excessive conduction and radiation heat gain during the day time. Vapor barriers must be applied to the outside surface of insulation to prevent moisture built-up within it, and to eliminate condensation problems. Fig. 4.5 shows an external wall section of a typical south-facing residential unit. Interior partitions are better made of light construction such as wood studs with plywood sheathing than with high mass such as concrete or bricks as is usually the case in Taiwan (see Chapter II, section 1.3).

Double glazing is preferable to single glazing because of its reduction of conduction and radiation loads. External louvered blinds can be used to efficiently reduce the radiation heat gain by reflection. Givoni pointed out that rough external wall surfaces have lower surface resistance values than smooth ones, i.e. the conduction load is higher with a rough surface. This load is negligible, though if the walls are light-colored and well insulated.

The roof should be covered with white pebbles, and the wall must be painted light color to reflect the solar gain. In urban areas such as Taipei, air pollution can easily darken the light-colored wall, and some washable surface material, therefore, such as ceramic might well be used.

5.0 SEALED BUILDING

The weatherstripping all joints as shown in Fig. 4.6 makes the building well sealed. The sensible as well as latent cooling load from infiltration can thereby be reduced.

6.0 VENTILATION

6.1 Ventilation Schedule

According to the Aren's bioclimatic chart analysis, we can construct a ventilation schedule on the M.A.H.W.D. chart. The requirement of different wind velocities are shown to cool the occupants inside the building (Fig. 3.6). In summer time, once the air temperature is above 82°F, the usual wind speeds required for cooling are always above 300 ft per minute (fpm) according to the bioclimatic chart analysis. During this time, the windows are preferably closed for two reasons:

- 1) 300 fpm wind speed is too high to be comfortable (Olgyay, Design with Climate p.20).
- 2) The outdoor excessive sensible and latent heat will greatly increase the overall cooling load in the internal space if the building is ventilated.

From the analysis above, the ventilation schedule on a M.A.H.W.D. chart (when the window should be opened for air flow) is shown in Fig 4.7.

EXTERIORS THRESHOLDS AND WEATHERSTRIPPING

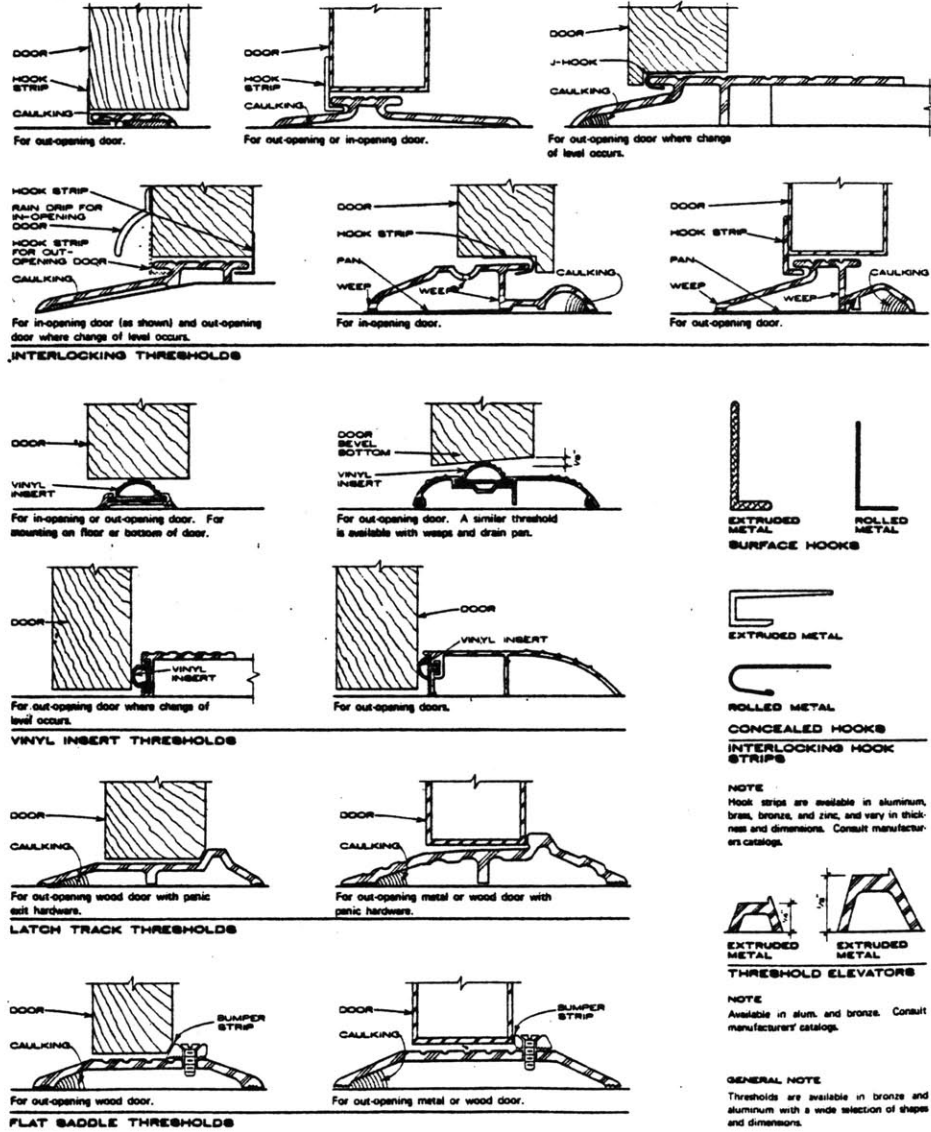


Fig. 4.6 Weatherstripping at Window and Door Frame

WEATHERSTRIPS FOR WINDOWS

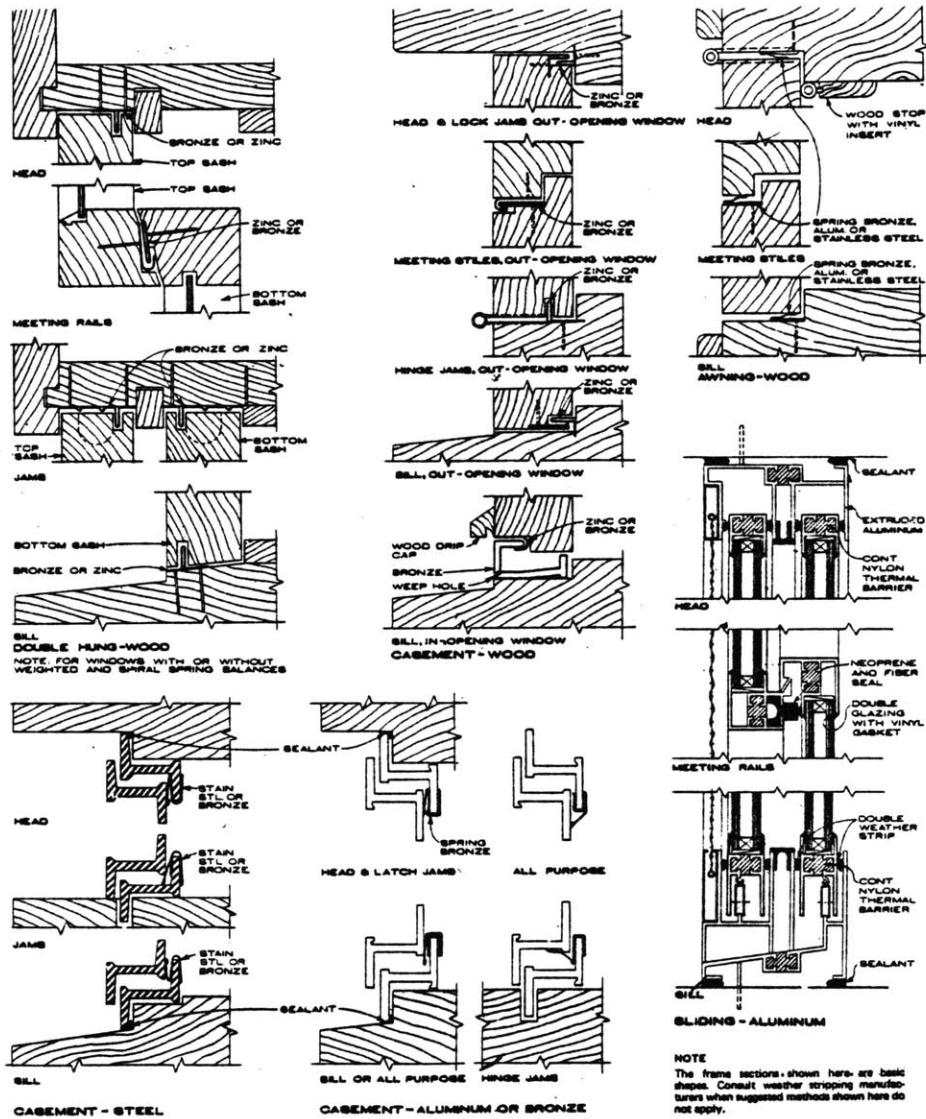


Fig. 4.6

[illegible]

82

MONTHLY AVERAGE HOURLY WEATHER DATA CHART													TAIPEI, TAIWAN R.O.C.	
	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC		
Tmax.	64.7 RH	69.3 RH	71.2 RH	77.2 RH	80.9 RH	84.9 RH	89.7 RH	89.7 RH	86.8 RH	80.6 RH	72.0 RH	61.9 RH	59.7 RH	
Tmin.	56.9 RH	57.2 RH	60.4 RH	67.1 RH	71.7 RH	79.6 RH	77.9 RH	78.2 RH	75.7 RH	71.6 RH	64.8 RH	59.8 RH	55.2 RH	
Tave.	60.0 RH	60.5 RH	64.0 RH	71.4 RH	75.4 RH	80.4 RH	83.8 RH	83.4 RH	80.5 RH	75.9 RH	68.1 RH	62.8 RH	57.1 RH	
TIME:	T RH	F RH	DRY-BULB TEMPERATURES, R.H., & WIND SPEEDS											
0 - 1 AM	56.1 RH	58.4 RH	60.8 RH	68.4 RH	72.7 RH	76.9 RH	79.9 RH	79.4 RH	77.4 RH	72.9 RH	66.4 RH	61.0 RH	56.0 RH	
1 - 2 AM	57.8 RH	58.1 RH	61.3 RH	68.0 RH	72.2 RH	76.4 RH	79.9 RH	79.4 RH	77.4 RH	72.9 RH	66.4 RH	61.0 RH	56.0 RH	
2 - 3 AM	57.5 RH	57.7 RH	61.0 RH	67.7 RH	71.9 RH	76.1 RH	79.6 RH	79.1 RH	77.1 RH	72.6 RH	66.1 RH	60.7 RH	55.7 RH	
3 - 4 AM	57.3 RH	56.7 RH	60.0 RH	67.5 RH	71.7 RH	75.9 RH	79.4 RH	78.9 RH	76.9 RH	72.4 RH	65.9 RH	60.5 RH	55.5 RH	
4 - 5 AM	57.1 RH	57.4 RH	60.8 RH	67.5 RH	71.7 RH	75.9 RH	79.4 RH	78.9 RH	76.9 RH	72.4 RH	65.9 RH	60.5 RH	55.5 RH	
5 - 6 AM	57.0 RH	57.2 RH	60.4 RH	67.1 RH	71.3 RH	75.5 RH	79.0 RH	78.5 RH	76.5 RH	72.0 RH	65.5 RH	60.1 RH	55.1 RH	
6 - 7 AM	56.9 RH	58.1 RH	60.6 RH	67.3 RH	71.5 RH	75.7 RH	79.2 RH	78.7 RH	76.7 RH	72.2 RH	65.7 RH	60.3 RH	55.3 RH	
7 - 8 AM	57.7 RH	58.0 RH	61.9 RH	68.6 RH	72.8 RH	77.0 RH	80.5 RH	80.0 RH	78.0 RH	73.5 RH	67.0 RH	61.6 RH	56.6 RH	
8 - 9 AM	59.5 RH	62.1 RH	64.0 RH	70.7 RH	74.9 RH	79.1 RH	82.6 RH	82.1 RH	80.1 RH	75.6 RH	69.1 RH	63.7 RH	58.7 RH	
9 - 10 AM	61.5 RH	65.1 RH	66.0 RH	72.7 RH	76.9 RH	81.1 RH	84.6 RH	84.1 RH	82.1 RH	77.6 RH	71.1 RH	65.7 RH	60.7 RH	
10 - 11 AM	62.3 RH	64.8 RH	67.5 RH	74.2 RH	78.4 RH	82.6 RH	86.1 RH	85.6 RH	83.6 RH	79.1 RH	72.6 RH	67.2 RH	62.2 RH	
11 - 12 AM	64.5 RH	65.2 RH	68.9 RH	75.6 RH	79.8 RH	84.0 RH	87.5 RH	87.0 RH	85.0 RH	80.5 RH	74.0 RH	68.6 RH	63.6 RH	
12 - 1 PM	64.7 RH	65.0 RH	68.7 RH	75.4 RH	79.6 RH	83.8 RH	87.3 RH	86.8 RH	84.8 RH	80.3 RH	73.8 RH	68.4 RH	63.4 RH	
1 - 2 PM	64.4 RH	64.3 RH	68.0 RH	74.7 RH	78.9 RH	83.1 RH	86.6 RH	86.1 RH	84.1 RH	79.6 RH	73.1 RH	67.7 RH	62.7 RH	
2 - 3 PM	63.7 RH	63.9 RH	67.6 RH	74.3 RH	78.5 RH	82.7 RH	86.2 RH	85.7 RH	83.7 RH	79.2 RH	72.7 RH	67.3 RH	62.3 RH	
3 - 4 PM	62.8 RH	62.9 RH	66.6 RH	73.3 RH	77.5 RH	81.7 RH	85.2 RH	84.7 RH	82.7 RH	78.2 RH	71.7 RH	66.3 RH	61.3 RH	
4 - 5 PM	61.7 RH	61.3 RH	65.0 RH	71.7 RH	75.9 RH	80.1 RH	83.6 RH	83.1 RH	81.1 RH	76.6 RH	70.1 RH	64.7 RH	59.7 RH	
5 - 6 PM	60.7 RH	60.5 RH	64.2 RH	70.7 RH	74.9 RH	79.1 RH	82.6 RH	82.1 RH	80.1 RH	75.6 RH	69.1 RH	63.7 RH	58.7 RH	
6 - 7 PM	60.2 RH	59.9 RH	63.6 RH	70.1 RH	74.3 RH	78.5 RH	82.0 RH	81.5 RH	79.5 RH	75.0 RH	68.5 RH	63.1 RH	58.1 RH	
7 - 8 PM	59.8 RH	59.4 RH	63.1 RH	69.6 RH	73.8 RH	78.0 RH	81.5 RH	81.0 RH	79.0 RH	74.5 RH	68.0 RH	62.6 RH	57.6 RH	
8 - 9 PM	59.3 RH	59.1 RH	62.8 RH	69.3 RH	73.5 RH	77.7 RH	81.2 RH	80.7 RH	78.7 RH	74.2 RH	67.7 RH	62.3 RH	57.3 RH	
9 - 10 PM	58.9 RH	58.9 RH	62.6 RH	69.1 RH	73.3 RH	77.5 RH	81.0 RH	80.5 RH	78.5 RH	74.0 RH	67.5 RH	62.1 RH	57.1 RH	
10 - 11 PM	58.6 RH	58.6 RH	62.3 RH	68.8 RH	73.0 RH	77.2 RH	80.7 RH	80.2 RH	78.2 RH	73.7 RH	67.2 RH	61.8 RH	56.8 RH	
11 - 12 PM	58.3 RH	58.3 RH	62.0 RH	68.5 RH	72.7 RH	76.9 RH	80.4 RH	79.9 RH	77.9 RH	73.4 RH	66.9 RH	61.5 RH	56.5 RH	

Fig. 4.7 Ventilation Schedule on A M.A.H.W.D. Chart

6.2 Wind Roses and Design Solutions

From the analysis of a regional wind rose for Taipei (see Fig. 3.19), we can conclude that in the spring, fall, and winter the prevailing wind is from east while in the summer it is from south, south south east (S.S.E.) and east south east (E.S.E.). The ventilation schedule tells us that we need ventilation for over half the year.

Therefore, the E., S., S.S.E., and E.S.E. winds must be induced or allowed to natural ventialetion in the spring, fall, and during summer nights.

Since the building's orientation in the city is uncertain, the four examples of wind inducing devices, correlated with the shading devices for facades facing different orientations are discussed below.

6.2.1 Main Facades Facing North and South

Summer winds can easily be used for natural ventialation directly, but spring and fall winds must be redirected by casement windows which create a positive pressure on the south facade and negative pressure on the north (see Chapter II,Section 2.4). The railing along the 1m wide balcony can also be provided with 45 vertical fins to help deflect the east prevailing wind into the building (based on the same principle), in the spring and fall. Fig. 4.8 shows the sunshading and wind-inducing design for the typical unit plan facing north and south.

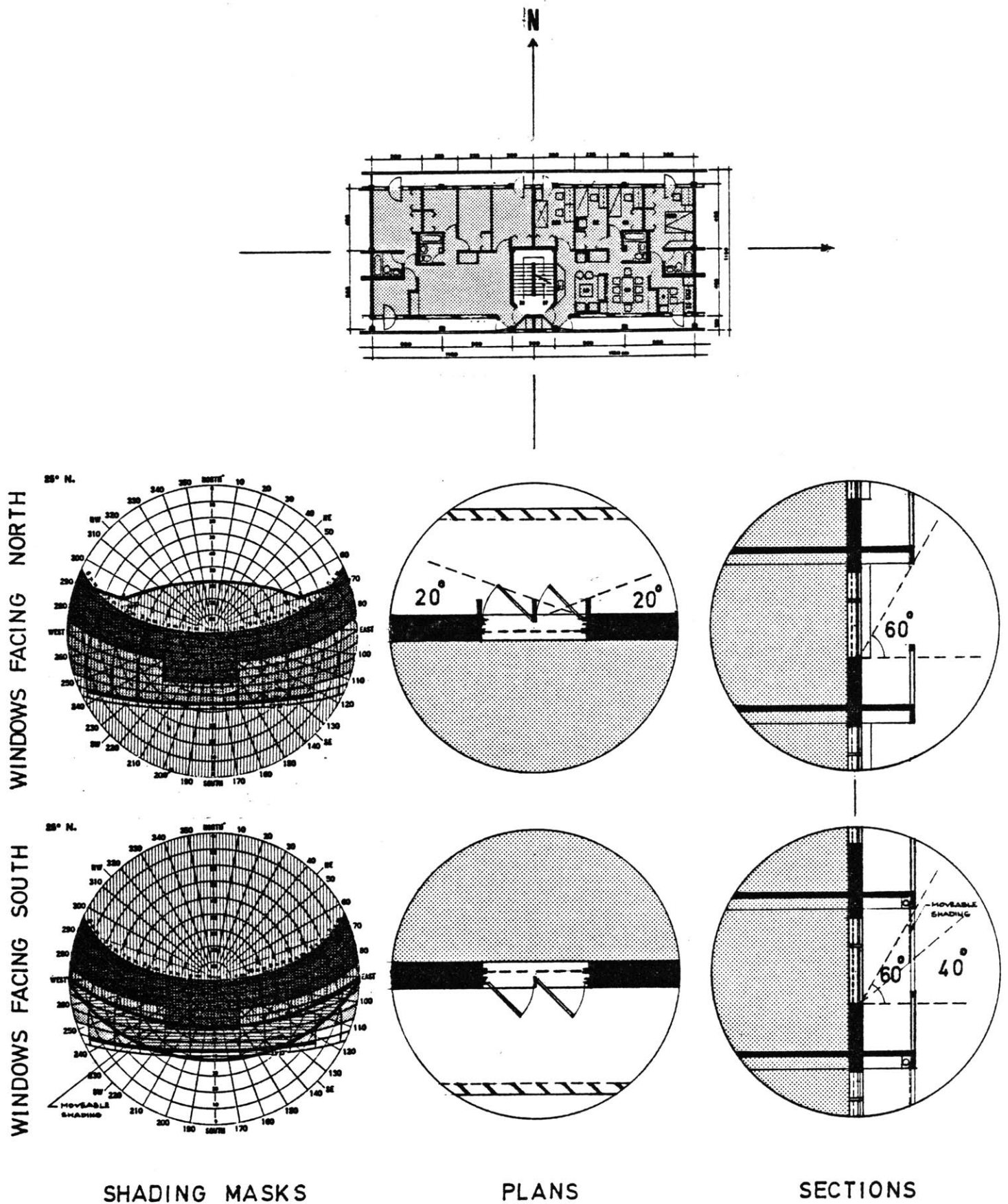


Fig. 4.8 Sunshading and Wind-inducing Design for North-South Orientated Buildings

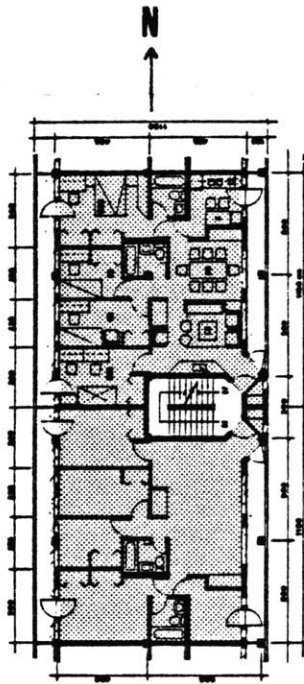
6.2.2 Main Facades Facing East and West

Two solutions can be introduced on the east side wall with two differently oriented vertical fins for sun control. Fig. 4.9 shows the optimal sun shading fins in east and west wall, once the shading devices are constructed, the spring and fall winds can still penetrate the building, while the summer winds cannot. In this case the vertical fins in the east balcony railing are used as the major wind inducing devices and louvers with movable insulation are applied below the window sill on the east side wall for wind induction (see Fig. 4.10).

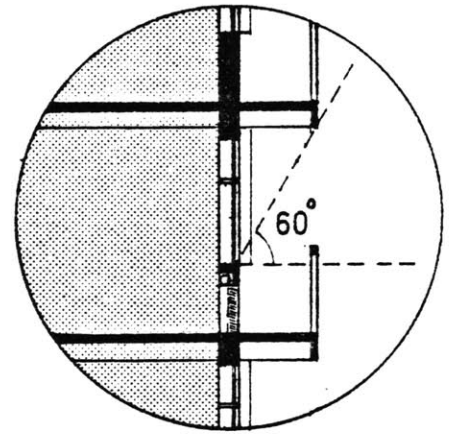
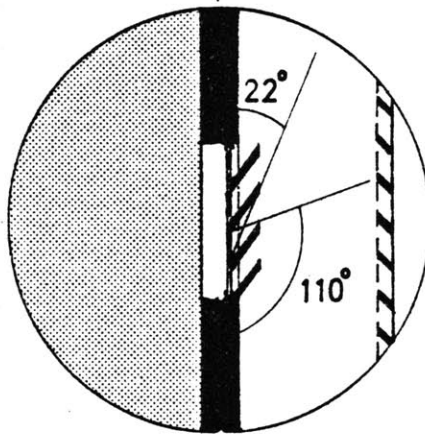
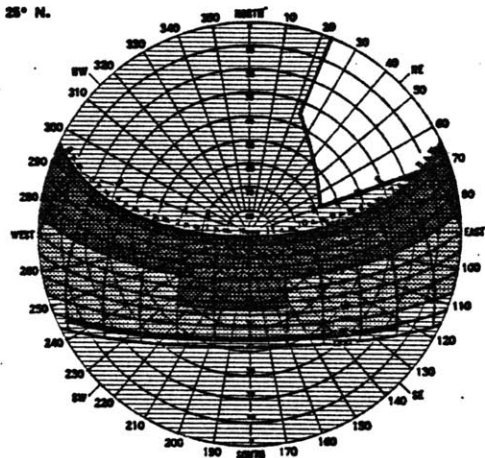
Fig. 4.11 shows a different sun shading devices on east side windows which cannot control the sun as efficiently as vertical fins shown in Fig. 4.9. Although it loses a little bit of over-all sun control, it is, nonetheless, a very good wind inducing device. In spring and fall winds can enter directly. In summer, the winds can be induced inside with the same principle as discussed before. It is always advisable to have oblique vertical fins in the railing to help induce winds in each case.

6.2.3 Main Facades Facing North-West and South-East

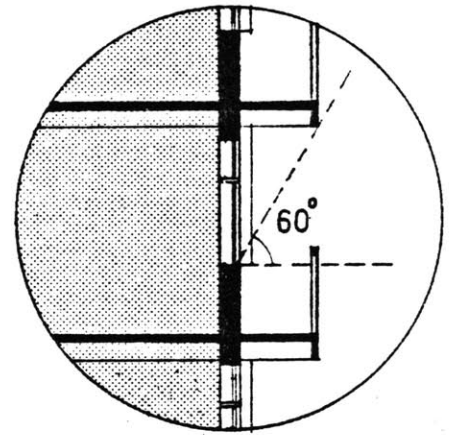
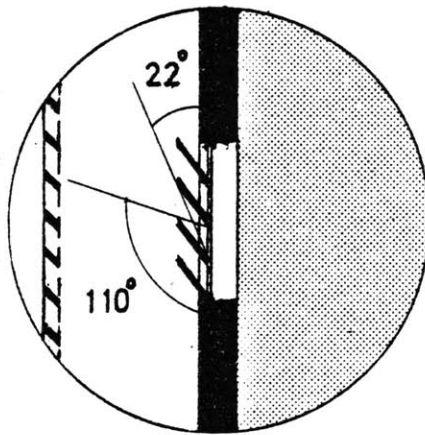
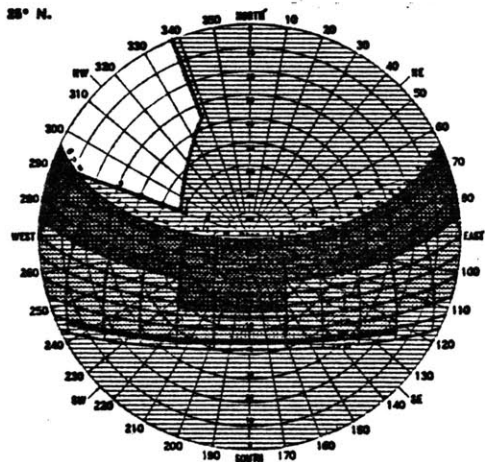
After sunshading vertical fins are constructed for this unit as shown in Fig. 4.12, summer winds can provide natural ventilation. Spring and fall winds (from E.) must be induced by vertical fins in the railing and cross-ventilation can be



WINDOWS FACING EAST



WINDOWS FACING WEST



SHADING MASKS

PLANS

SECTIONS

Fig. 4.9 Sunshading and Wind-inducing Design for East-West Orientated Buildings

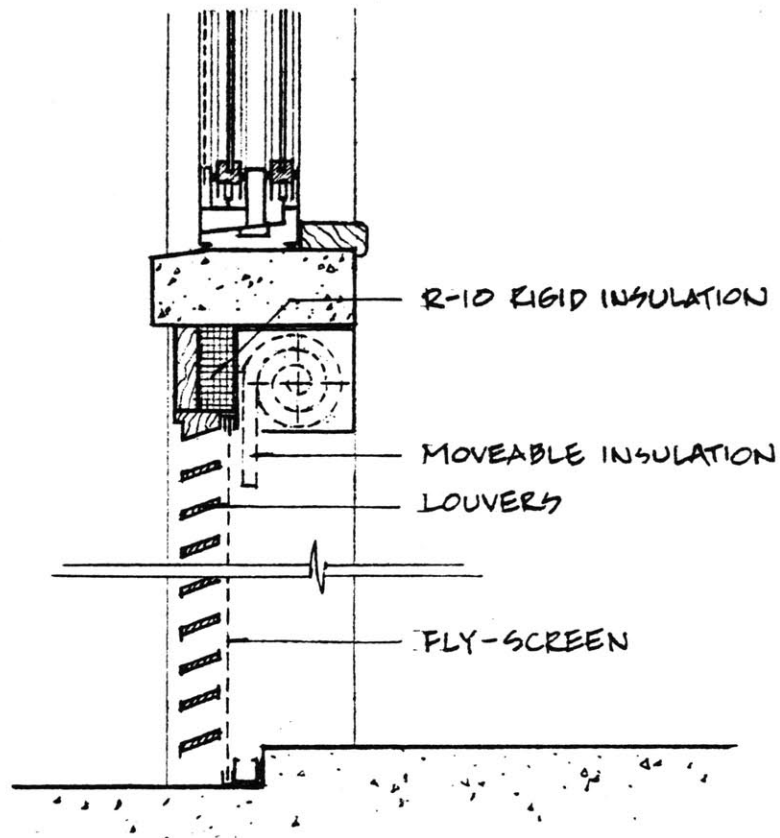


Fig. 4.10 Movable Insulation with Louvers below The Window Sill

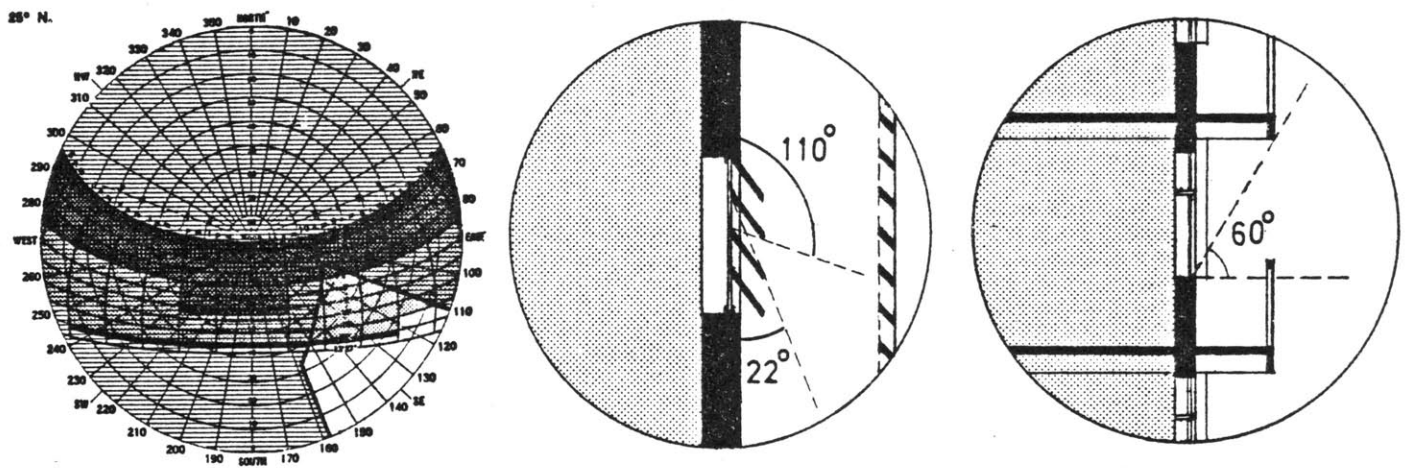


Fig. 4.11 A Second Option for Sunshading and Wind-Inducing Design in East-West Orientated Buildings

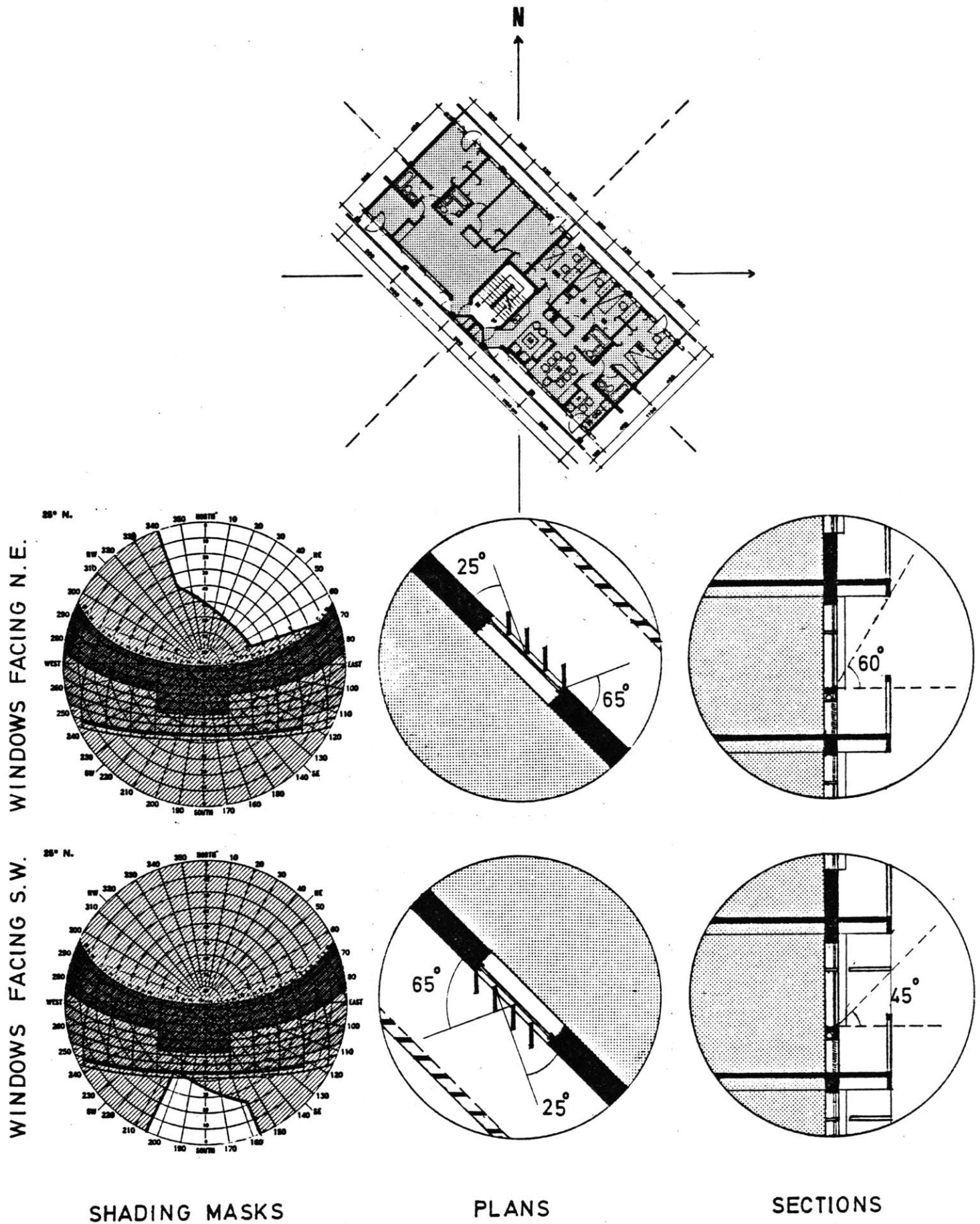


Fig. 4.12 Sunshading and Wind-Inducing Design for NE-SW Orientated Buildings

accomplished through the louvers with moveable insulation below the window sill (see Fig. 4.10).

6.2.4 Main Facades Facing North-West and South-East

The same problem occurs due to the east wind which cannot ventilate the building with the optimal sunshading devices (Fig. 4.13). The same solution as above which uses vertical fins in the railing and louvers with movable insulation under the window sill, must also be applied here.

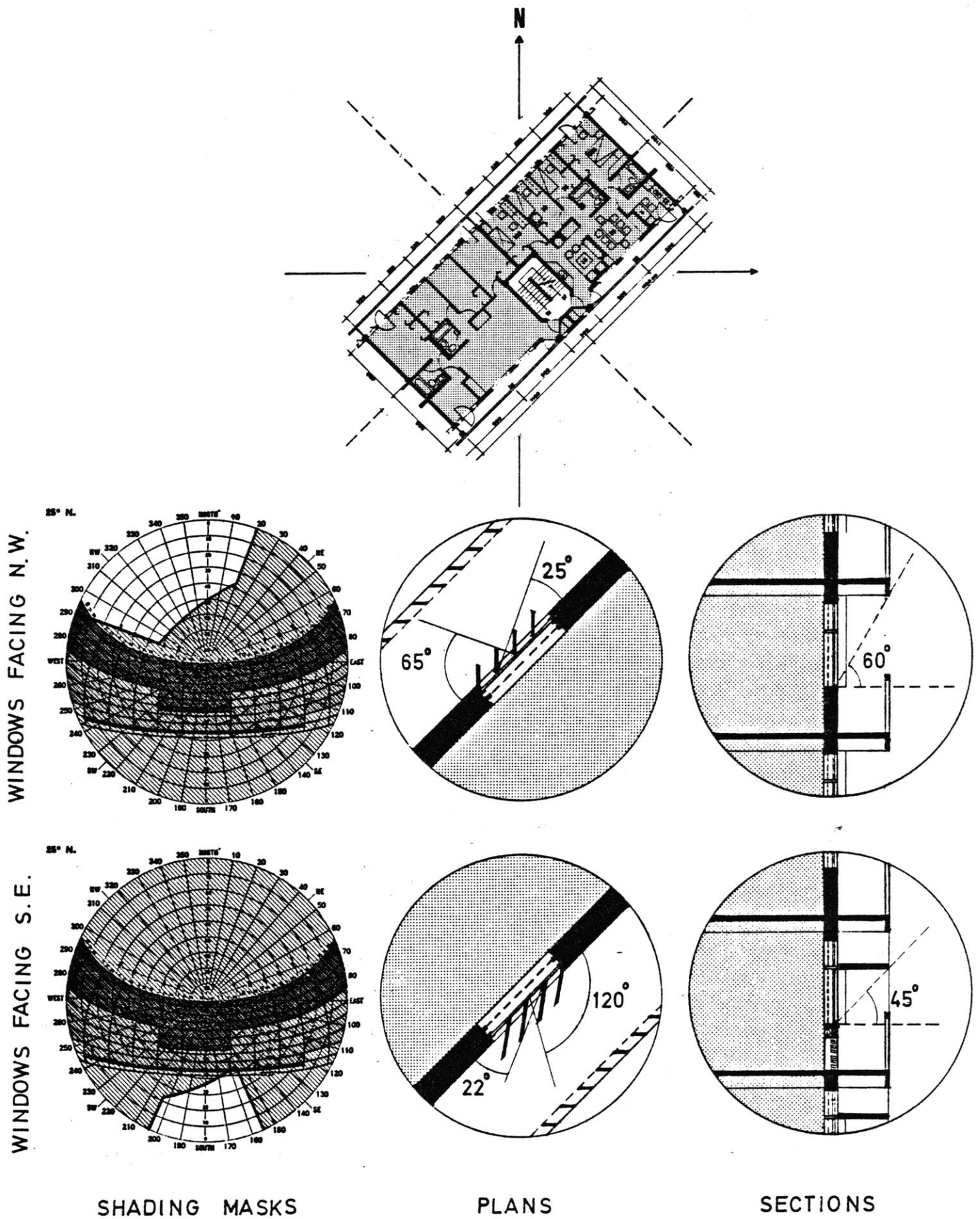


Fig. 4.13 Sunshading and Wind-inducing Design for NW-SE Orientated Buildings

REFERENCES FOR CHAPTER IV

1. B., Givoni, "Man, Climate and Architecture", 2nd ed., (Applied Science Publishers, Ltd., London, 1976), chapter 7.
2. E., Arens, "A New Bioclimatic Chart for Passive Solar Design", Proceedings of the American Section of the Internal Solar Energy Society, Vol. 9.2, October 19-22, 1980.

CHAPTER V

LOW COST COOLING USING ANNUAL STORAGE

1.0 INTRODUCTION

Of all the basic climate types, hot, humid climates are the most difficult ones to deal with passively. This difficulty is due to the high air temperatures and high humidities during the summer season. An analysis of both psychrometric and bioclimatic charts reveals that passive cooling techniques can not be used to cool during the daytime in July and August in such a climate region as Taipei.

In real applications, when weather conditions require at least a 300 ft/min wind speed to maintain comfort, the windows of a structure are better left closed as discussed in chapter IV. When ventilation is not appropriate, conventional dehumidifiers or air conditioning systems must be applied to reduce the sensible and latent loads in order to maintain comfort.

In this section, an alternative low-cost cooling technique will be introduced to replace the high-cost conventional dehumidifier or air-conditioning systems.

2.0 SUMMER COOLING LOAD

The summer cooling load for a 6-person extended family unit of the aforementioned multifamily residences facing

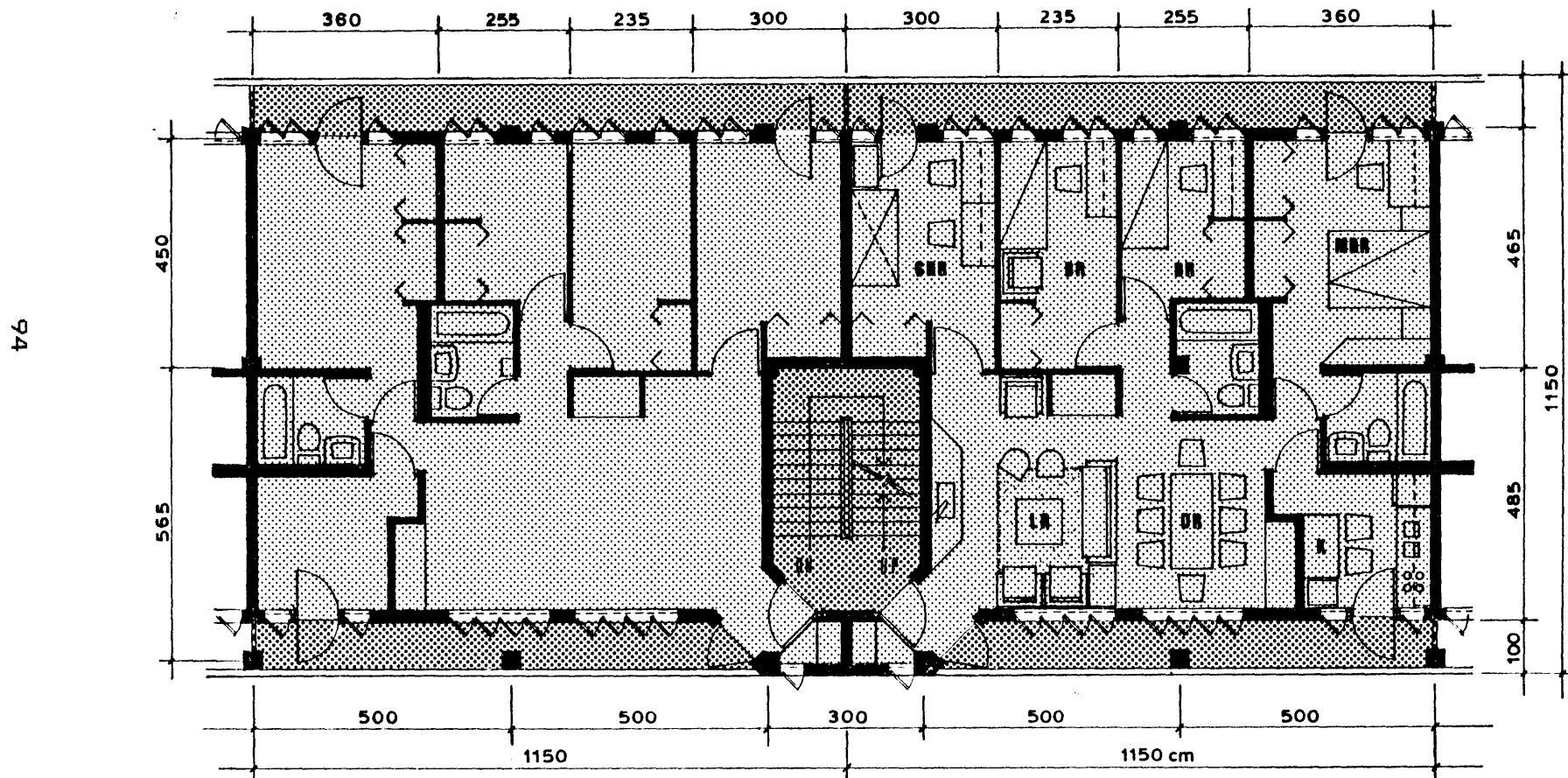


Fig. 5.1 Typical Unit Plan Facing South and Its Summer Load in Taipei

south or north (Fig. 5.1) or (fig. 4.18) are calculated in Appendix , and shown in Fig. 5.1.

The external heat gains are minimized by several means: 1) double glazing, 2) external louvered blinds, 3) R-11 opaque walls (Fig. 4.15), 4) R-10 insulated doors (Fig. 4.16), 5) direct sun shading. 6) sealed window or door frame (Fig. 4.16), 7) light color surfaces. The cooling load referred to only accounts for the periods when the windows are closed. That is to say, the wind is insufficient or inappropriate, during these periods, to be used as a cooling source. The loads must be extracted by a properly sized mechanical system, such as a conventional dehumidifier, air-conditioning system, or, better still, the new system which is discussed below.

3.0 HEAT PUMPS

A heat pump is a device which extracts heat energy from a source at a low temperature and adds this extracted energy to a source at a high temperature. If charge and discharge sources can be found, the heat pump applications can be worthwhile because the amount of energy required to operate this system is only a fraction of the total heating or cooling energy provided.

Heat pumps can be used to heat or cool the building, or to serve as a domestic hot water (DHW) heater. They have been in wide-spread case in Japan, the United States, and other countries in the world due to the convenience of

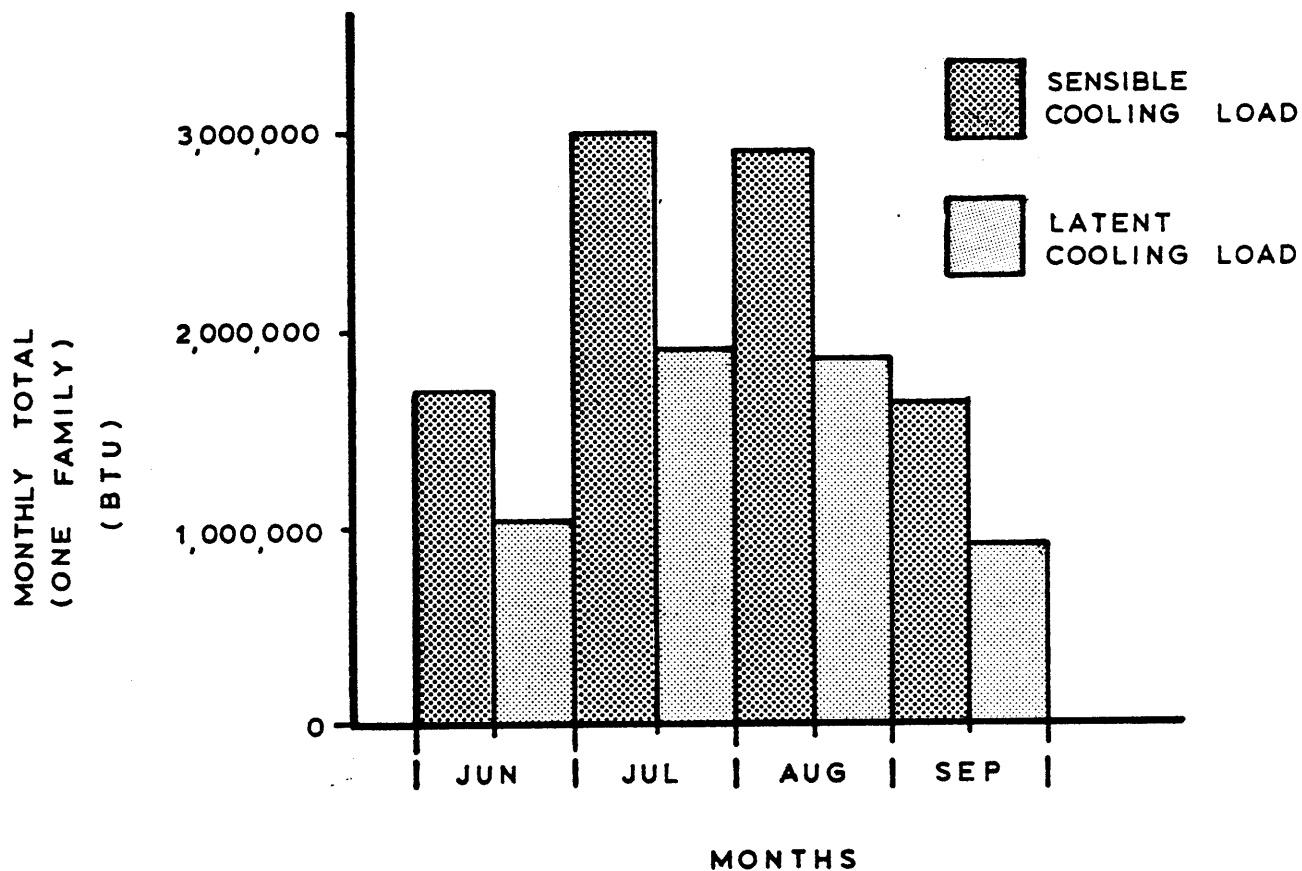


Fig. 5.1 Typical Unit Plan Facing South and Its Summer Load in Taipei

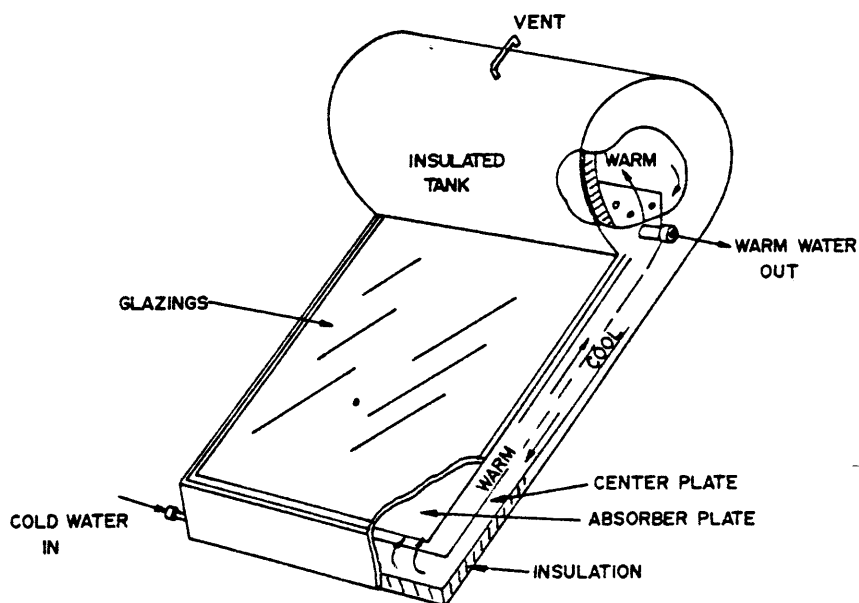


Fig. 5.2 Flat-Plate Thermosiphoning Solar Heater

switching to either heating or cooling modes in the building, and due to their competitive initial cost, long term durability, easy installation, and the low operating costs.

The ratio of energy (heat) extracted (Q) to operating energy input (q) is the basic measure of the effectiveness of a heat pump. This ratio is named "COP" (coefficient of performance), i.e. $COP = Q/q$. To be effective, a heat pump must have a COP greater than 1, and the higher the COP the more effective it will be, and the more costs will be saved.

In general, the heat pump function, during the cooling mode, produces higher temperature energy as a by-product, and this energy is rejected as waste. During the heating mode, the by-product of lower temperature energy is rejected as waste. If the so called "waste" energy were fully utilized as a "useful" energy, the COP of a heat pump could be imagined as having twice its previous value.

In hot, humid climates, latent loads are as serious a problem as sensible loads. Although desiccant dehumidification techniques may appear to be passive in removing latent loads, they will eventually increase the sensible load (as discussed in Chapter II). Heat pumps show their superiority in this case because they can remove latent as well as sensible loads.

4.0 HEAT PUMP, DHW, AND SEASONAL COOLING RESERVIOR

4.1 Domestic Hot Water Heater

There are many different ways to heat DHW for the occupants, such as propane heaters, electrical heaters, flat plate solar collectors, or heat pumps. The most energy consumptive device among this list is the electrical heater, second, the propane heater; although they are both widely used. The performance of flat plate solar collectors, whether by active or passive means (Thermo-siphon principle shown in Fig. 5.2), have drawn much attention recently. However, they depend significantly on the direct solar heat gains. Difuse solar radiation in cloudy days can contribute very little in water heating due to the temperatures required. A back-up system which uses electricity or gas must accompany the collector for cloudy days. In a region which has a high percent of cloudy days, and a high precipitation rate like Taipei, solar collectors have a real limitation. Heat pump DHW heaters, though they use electric energy, are definitely superior to solar water heaters in this case because of the lower initial cost and a constant performance independent of cloudiness. It is also a better choice than an electric or gas heater due to its much lower operating energy input. Akridge pointed out that recently marketed heat pump DHW heaters require only 35-50% as much energy input as required by conventional electric DHW heater and can be purchased for \$550-650 U.S. dollars.

Since the heat pump DHW units are very small and easily

moved, it is possible to locate the units within the occupied spaces. This economy makes it possible for the unit to not only meet the DHW needs, but also provide sensible and latent cooling as a side benefit.

In the summer, the air in the room needs to be cooled and dried. An air to water heat pump is useful and takes the energy removed from the space being cooled and add this energy to DHW. In the spring, fall, and winter seasons, however, the air does not need to be cooled or dried, and a water to water heat pump which extracts the heat removed from a water resource and add this heat to the DHW is useful. If the cooled water can be stored up gradually during these seasons, a huge cooling resource can be formed at the end of spring which can then be used as a "free-gift" sensible cooling medium in summer.

4.2 Seasonal Cooling Reservoir and Sensible Cooling

4.2.1 Seasonal Cooling Reservoir

The concept of coupling seasonal cooling reservoir with a water to water heat pump DHW heater shows its great merit in storing the by-product "waste" cooled energy and utilizing this energy to cool the building in the summer. The most critical problems with this application are how this "cooled" energy can be stored and kept cooled and how big volume this reservoir should be. One could imagine this reservoir would be very large because the water to water heat pump DHW heater keeps extracting heat from this

reservoir, and if the reservoir is not big enough, very soon it will freeze and thereby tremendously reduce the heat pump COP.

4.2.2 Reservoir Material

The material of this reservoir should have a high heat capacity in order to reduce the reservoir volume. It should also be easily accessible and cheap for the sake of economic feasibility. In addition, it should have a low deterioration and corrosion rate to insure durability.

Akridge experimented with water pipes buried underground and used the underground soil as a cooling reservoir. This approach proved successful in Georgia with insulation on the ground surface due to the low mean ground temperatures. This strategy, however, can not be applied to Taipei where the mean ground temperature is too high and the land usage is very limited due to extremely high land price resulting from a high population density. Some phase change materials have very high heat capacity. But, unfortunately, they are too expensive for this use. Among many easily accessible materials such as rock, soil, sand, etc., water has the highest heat capacity (66 Btu/cuft-°F). Water also has high conductivity which means it can transfer energy effectively. Moreover, it is the cheapest material in hot, humid climate. The low deterioration and corrosion rate means that it is easy to operate and maintain. From this comparison, it is concluded that the water is the best

material for the seasonal cooling reservoir mentioned above.

4.2.3 Reservoir's Volume and Location

The volume of this cooling reservoir (tank) is determined by the extraction rate of DHW heat pump heater. For the two units of typical 6-person family design in this thesis, the total volume of a cooling reservoir is about 11,310 cuft, approximately 40% of two typical unit's construction volume (including the circulation area). This volume is calculated in Appendix B .

The location of this reservoir is preferably underground. The soil which surrounds it can be used as additional insulation and the construction of this reservoir can serve as a floating foundation of the building (Fig.5.3). Thus, without much increase in initial construction cost, this floating foundation cooling reservoir can provide sensible cooling for two families in summer.

4.2.4 The Amount of Insulation

Insulation must be added to the cooling reservoir in order to reduce heat gains from surroundings. The amount of insulation is approximately R-50 in ceiling and in walls. If horizontally extended R-15 insulation is applied to the edge of the building as shown in Fig. 5.4, the insulation in the walls can be reduced from R-50 to R-20. It is because the mass of soil can provide another R-15 insulation for the

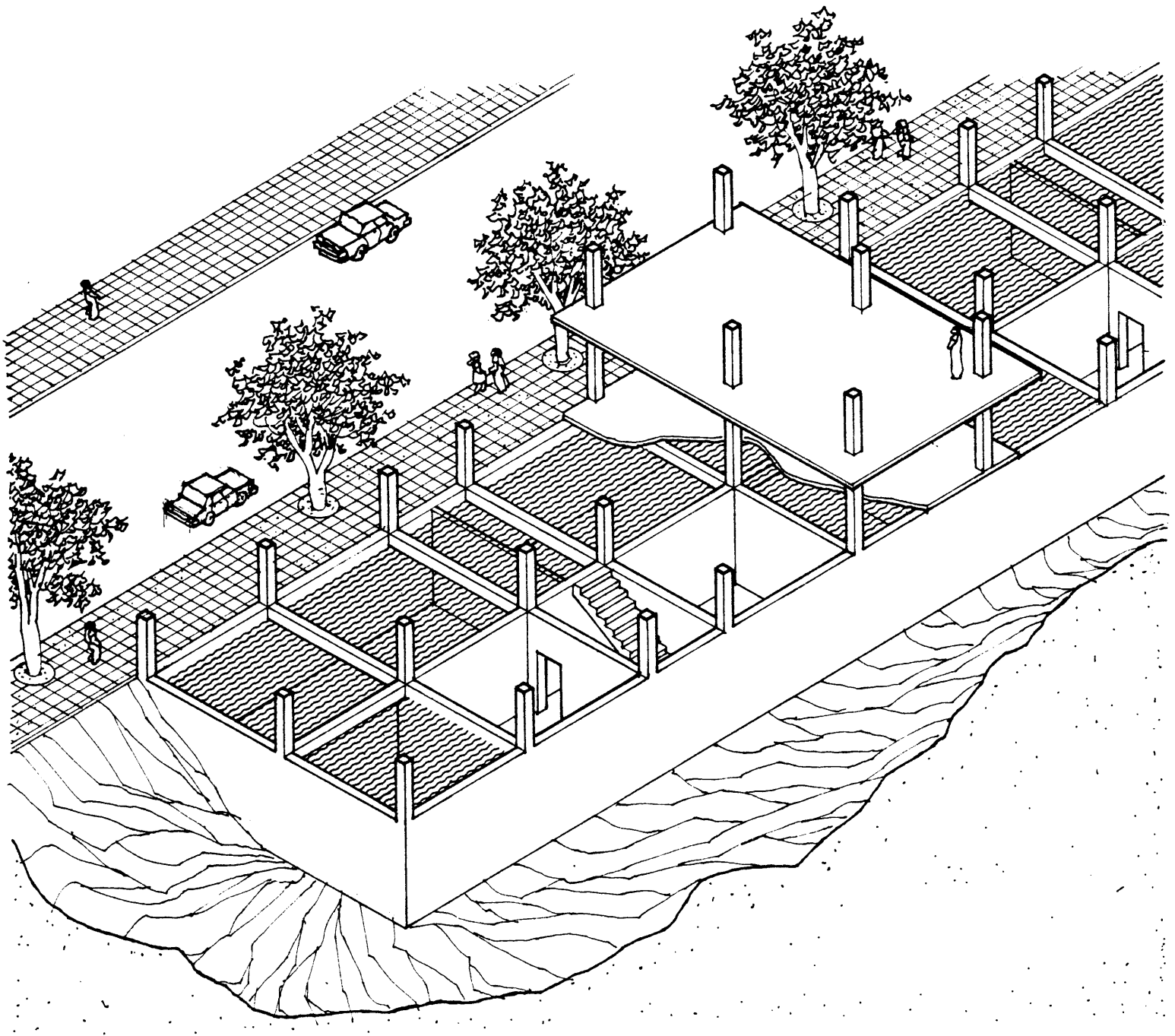


Fig. 5.3 Cooling Reservoir as A Floating Foundation for Buildings

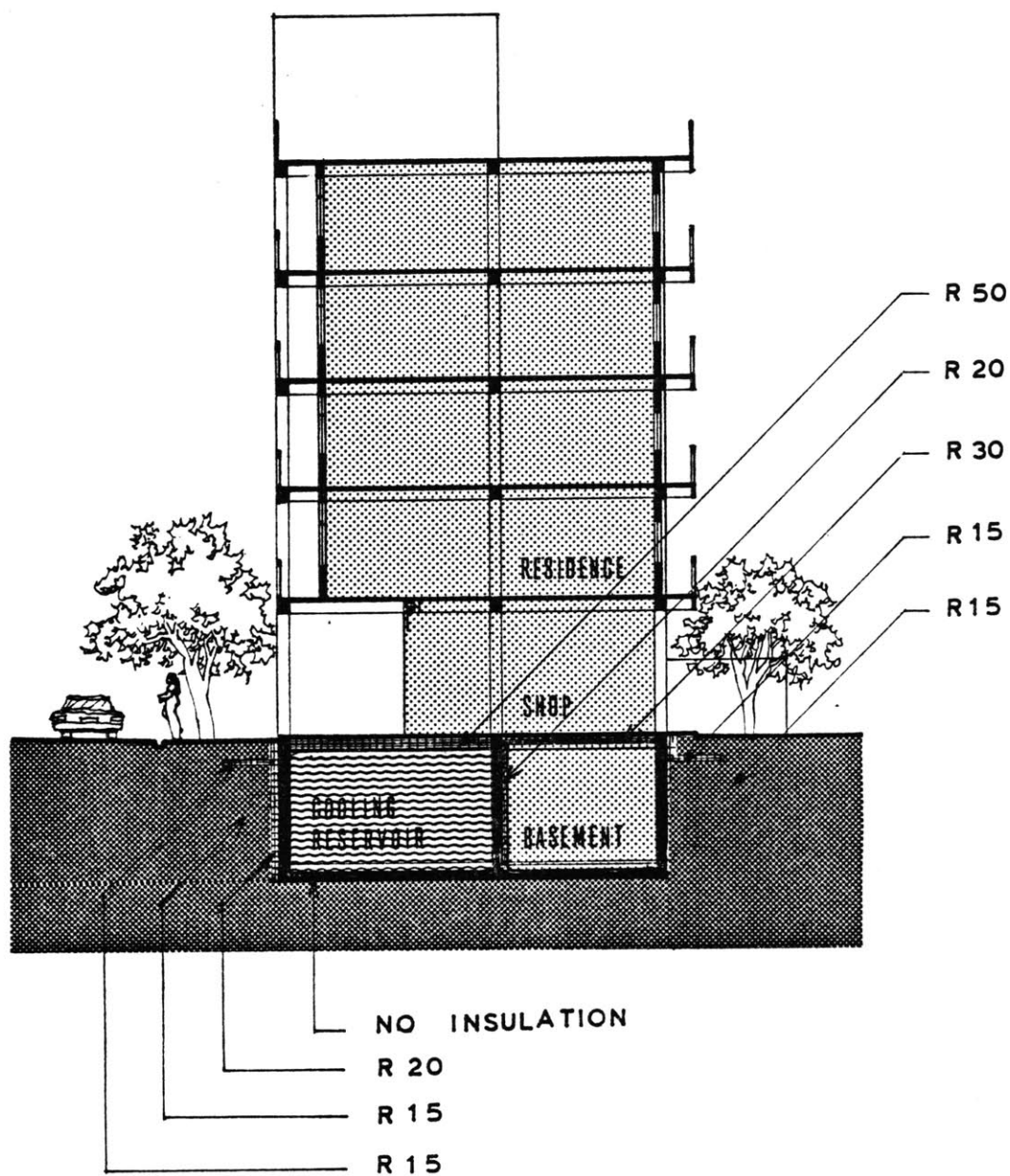


Fig. 5.4 The Insulation Effect of Soil Mass around The Reservoir

heat flux reduction from the external air temperature.

The bottom floor of the reservoir needs no insulation at all because of the low water table of Taipei which is 40 meters below ground surface. The thermal gradients of the soil mass between the bottom of the reservoir and the water table (see Fig. 5.5) can serve as a perfect insulation in reducing great heat gain from mean ground water temperature (73°F). The soil temperature next to the reservoir fluctuates directly according to the stored water temperature.

Details of insulation in ceiling and walls of the reservoir, and horizontally extended insulation at the edge are shown in Fig. 5.6.

4.2.5 Charge and Discharge Periods

If we keep extracting heat from the cooling reservoir for DHW needs from Oct.-May. It can be seen from the calculations in Appendix B that the total stored cooling potential at the end of these eight months can displace the sensible cooling load for the other four months (June-Sept.), whenever ventilation is an inappropriate cooling strategy.

Therefore, the charge periods should be from Oct. to May which theoretically reduce the water temperature in the cooling reservoir from 70°F to 40°F (below which the COP of the heat pump would decrease significantly). The discharge periods should be from July to Sept. which theoretically

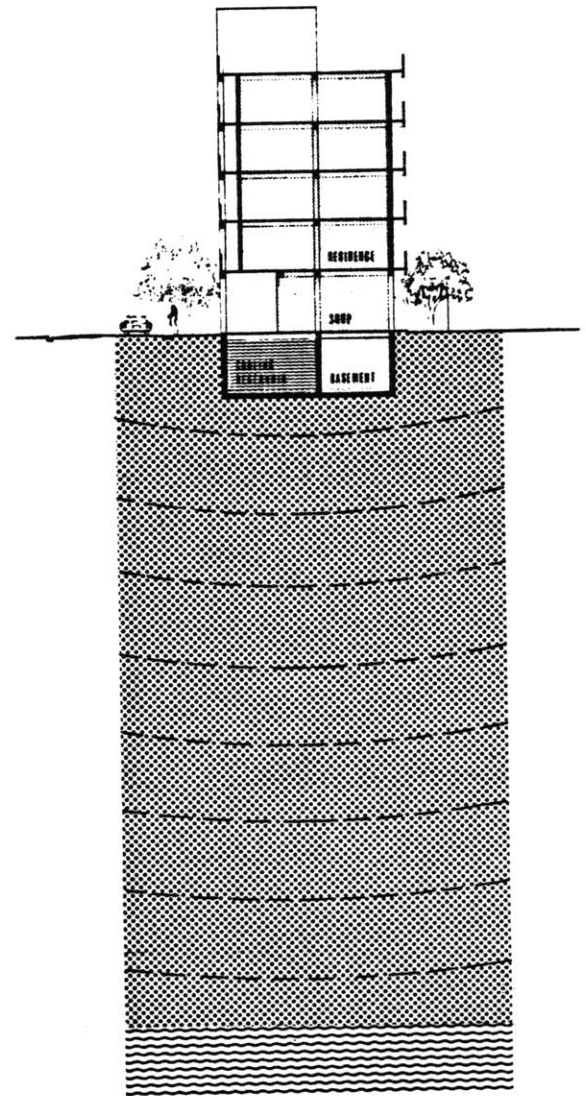


Fig. 5.5 The Insulation Effect of Soil Mass below The Reservoir

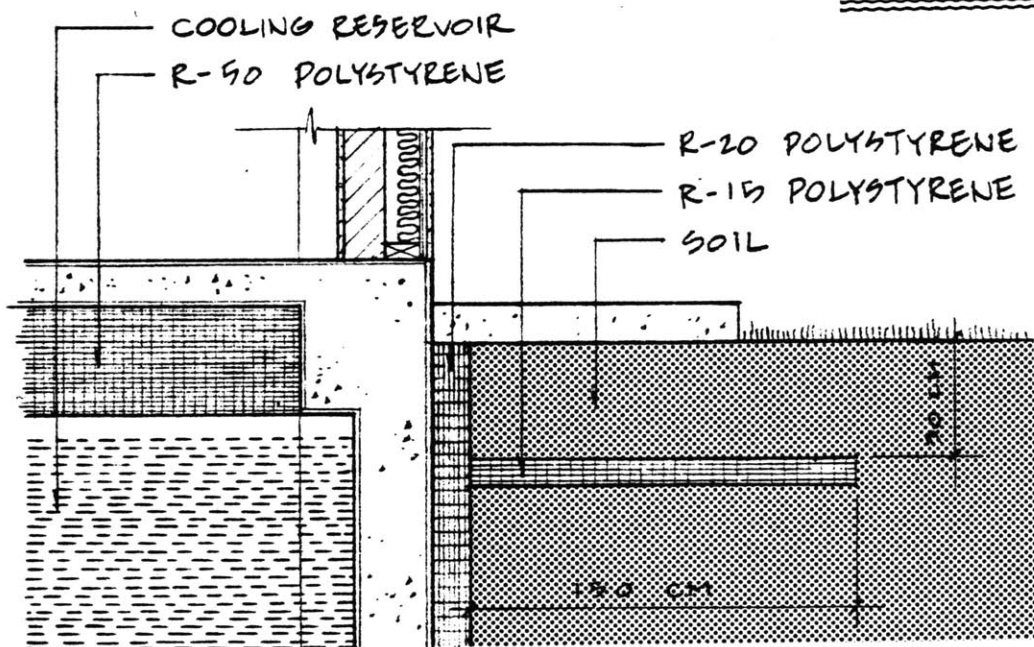


Fig. 5.6 Horizontally Extended Insulation Detail at The Building Edges

increase the water temperature from 40°F to 70 °F (above which the water cannot be used as a cooling medium). Thus, a yearly cycle of this system is established.

During the summer the water-to-water heat exchanger should be replaced by an air-to-water heat exchanger for the same heat pump DHW heater to reduce the latent cooling load of the apartment which is to be discussed later.

4.2.6 Sensible Cooling - Radiant Panel System

In recent years, radiant panel ceilings have received much attention and have been widely applied as a heating or cooling system in various types of buildings. ASHRAE proclaims that comfort levels in this system are better than those experienced in any other conditioning systems. It is because the cooling load in the building is treated directly by the large area of radiant heat sink.

This system usually consists of serpentine copper coils bonded to the ceilings. On a cooling cycle, the chilled water from the cooling reservoir is circulated through the ceiling coils. The ceiling panels, being cooled, serve as a radiation sink which absorbs sensible heat from the conditioned space.

The temperature of the chilled water circulated through the coils must be above the dew-point temperature of the conditioned space in order to prevent condensation problems. If the interior condition is 78°F with 50% RH as for a typical residence, the temperature of the circulated water

must be above 58°F which is the dew-point temperature of the space as read on a psychrometric chart (see Fig. 5.7). If the temperature of the supplied water is below 58 °F, a four-pipe system proposed by Obrecht as shown in Fig. 5.8 can be used to control condensation without sacrificing any cooling capacity.

The water flow rate will affect cooling potential of a low-mass radiant panel system. The higher the flow rate, the greater the cooling potential of the panel would be. This phenomenon is due to the low rise in outlet water temperature resulting in a lower mean water temperature which provides a higher cooling capacity. Obrecht indicates the water quantity circulated through radiant cooling panels is usually based on a 5°F rise in water temperature. For high-mass radiant panel system however, as tested by Akridge, the surface temperature of the panels is not influenced by a change in flow rates. This means that the water flow rate does not affect the cooling potential of this high-mass panel system.

Fig. 5.9 shows the detail of one type of radiant ceiling panels applied to the typical family design mentioned above. The distance between the pipes is determined by the cooling load and are sized by the method developed by Obrecht.

4.3 Heat Pump and Latent Cooling

In hot, humid climates the summer latent load is the

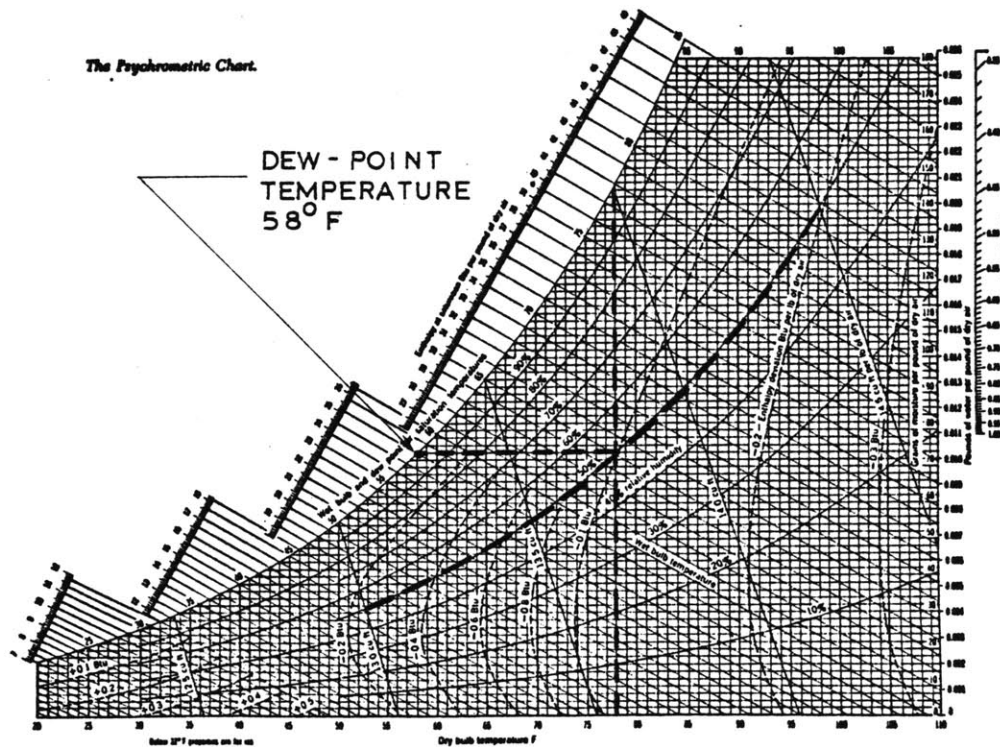


Fig. 5.7 The Dew-Point Temperature of A Typical Conditioned Space on A Psychrometric Chart

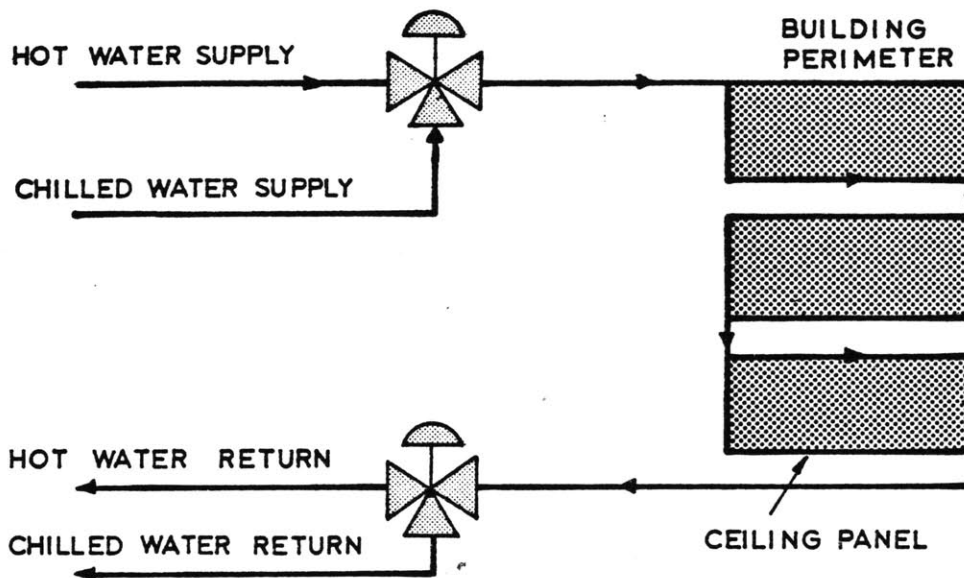


Fig. 5.8 Radiant Panel Condensation Control with A Four-Pipe System

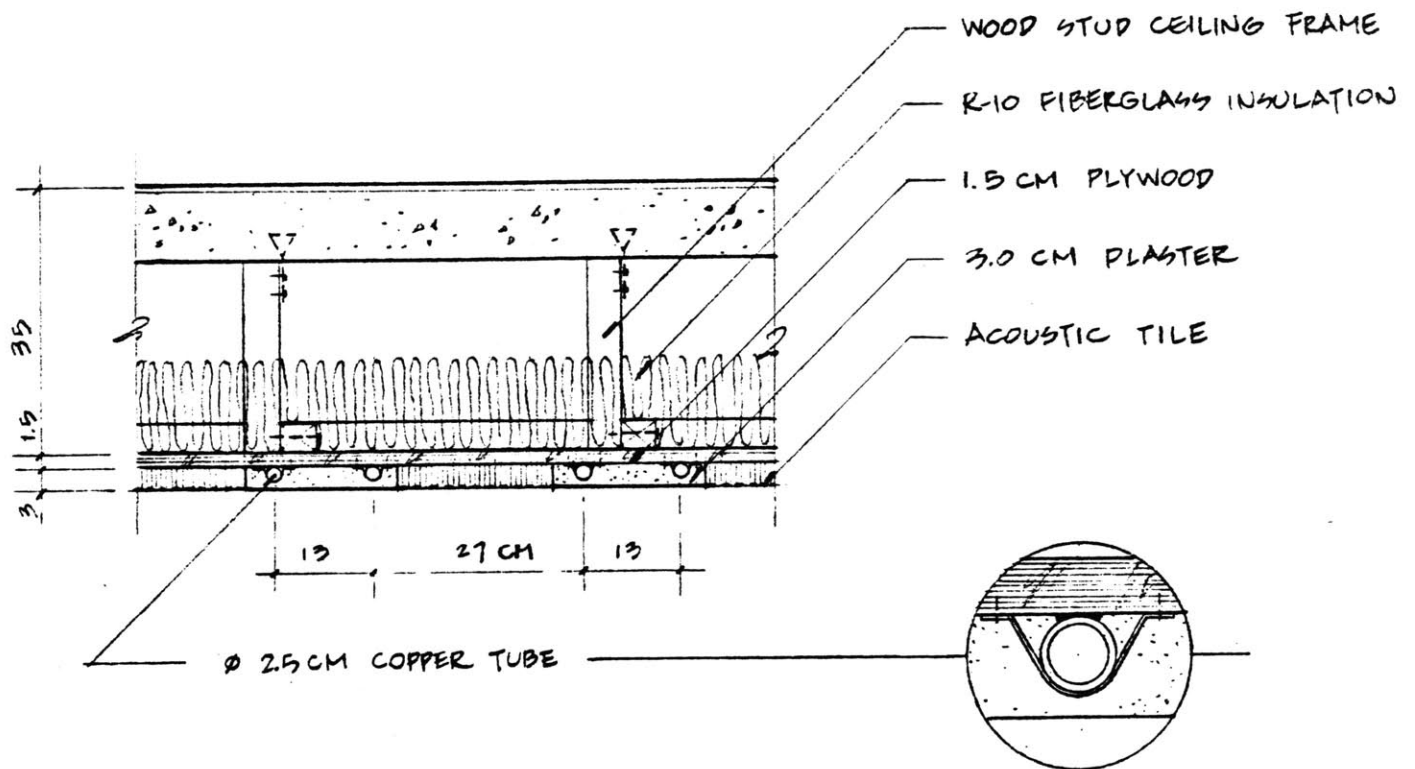


Fig. 5.9 Detail of Ceiling Panels for radiant Cooling

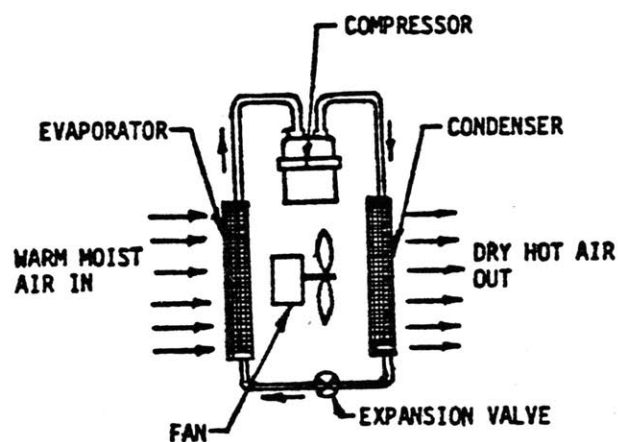


Fig. 5.10 Basic Components of A Conventional Dehumidifier

most difficult problem to deal with. Although the dessicant dehumidification technique may appear passive, it results in raising the sensible load and is not applicable as is mentioned above (Fig. 2.6), to hot humid climates.

A conventional mechanical dehumidifier is also very inefficient on account of high energy consumption (COP less than 1). It has the same problem of removing latent heat but increasing sensible heat. The heat removed as latent energy by the evaporator is added back to the outlet air at condenser, and thus the sensible load is increased. Fig. 5.10 shows the component layout and basic operating mode of conventional dehumidifiers.

Before discussing more effective techniques for removing latent load, the load itself must first be minimized by a well sealed building in terms of reducing the infiltration rate.

4.3.1 Heat pump DHW Heater and Dehumidification

A conventional water source heat pump has a higher COP than that of a dehumidifier and removes latent heat as well as sensible heat. The basic concept is that the heat removed from the air at the evaporator is carried away as waste by water circulated at the condenser to the discharge well (Fig. 5.11). A heat pump DHW heater with an air-water heat exchanger shows its great merit because the heat removed from the air is added to the DHW and as such is not wasted, but rather is fully utilized (Fig. 5.12).

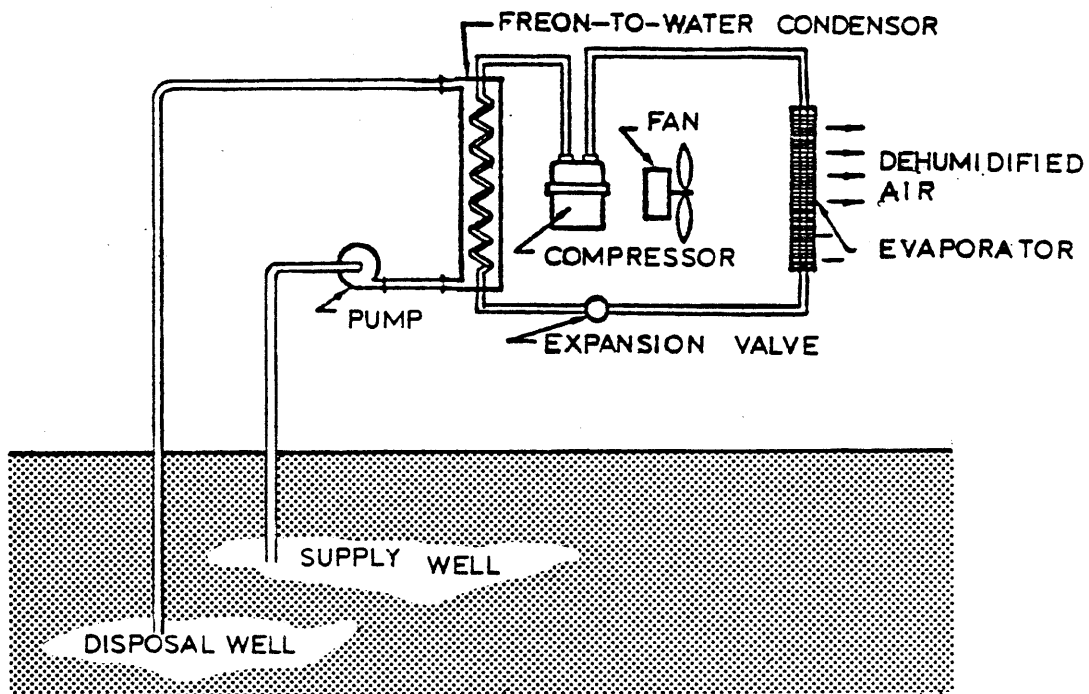


Fig. 5.11 Water Source Heat Pump

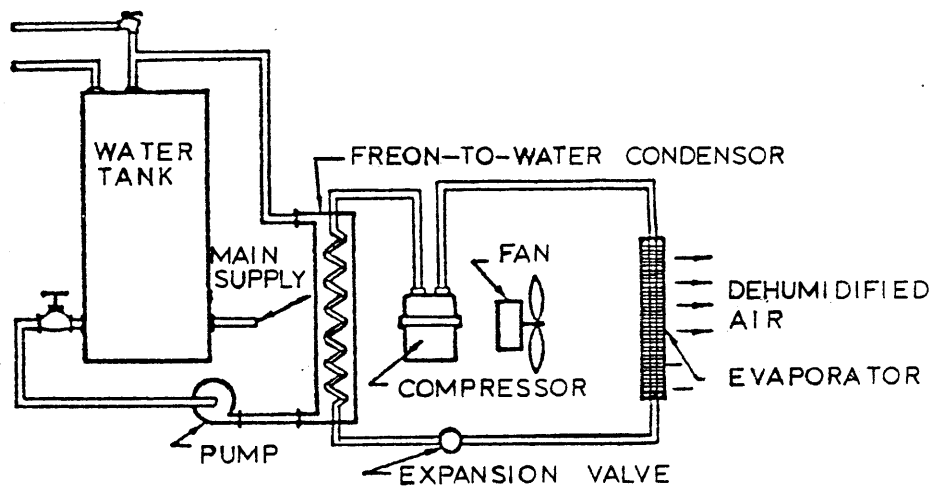


Fig. 5.12 Conventional Heat Pump DHW Heater

4.3.2 Run-around Coil

A run-around coil proposed by Akridge will increase the system's efficiency and its latent cooling capacity. Fig. 5.13 shows the basic run-around cycle. Water is circulated between two air-water heat exchangers located at the both sides of the mechanical system's evaporator (cooling coil). The first coil in which air is circulated is called the precool coil and the last one is called the reheat coil.

As air is induced and first cooled by the precool coil, the lesser amount of heat in the air is left for the system's cooling coil to carry, thus increasing the system's performance. After air is condensed and cooled at the evaporator, the sensible heat removed from the precool coil is added back to the chilled air at the reheat coil, thus decreasing the sensible cooling capacity and increasing the latent cooling capacity.

For the case in which sensible load can be carried by the cooling reservoir with the radiant panel cooling system, the run-around coil should be added to a heat pump DHW system to bias the system towards more latent cooling and less sensible cooling, thus again, improving the system's efficiency.

Figure 5.14 shows a heat pump DHW heater with run-around coil proposed by Akridge, the COP of this system

4.4 Overall Cooling Strategies

4.4.1 Heat Pump Operation Schedule

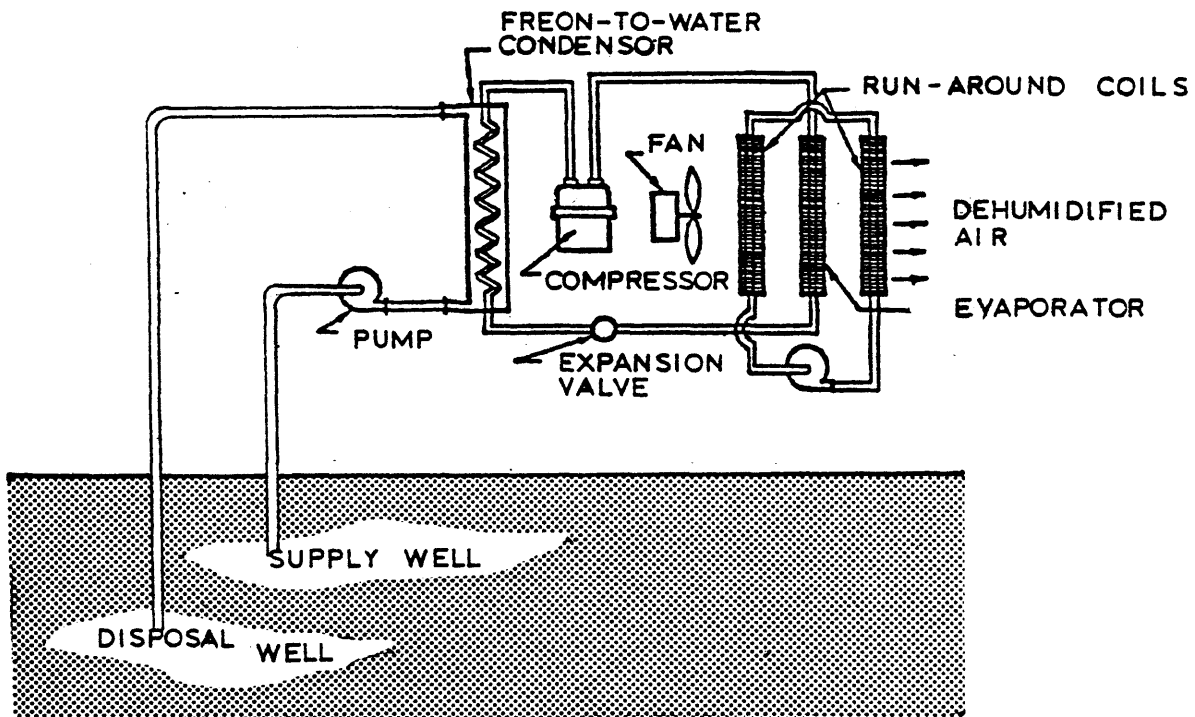


Fig. 5.13 Water Source Heat Pump with A Run-Around Coil

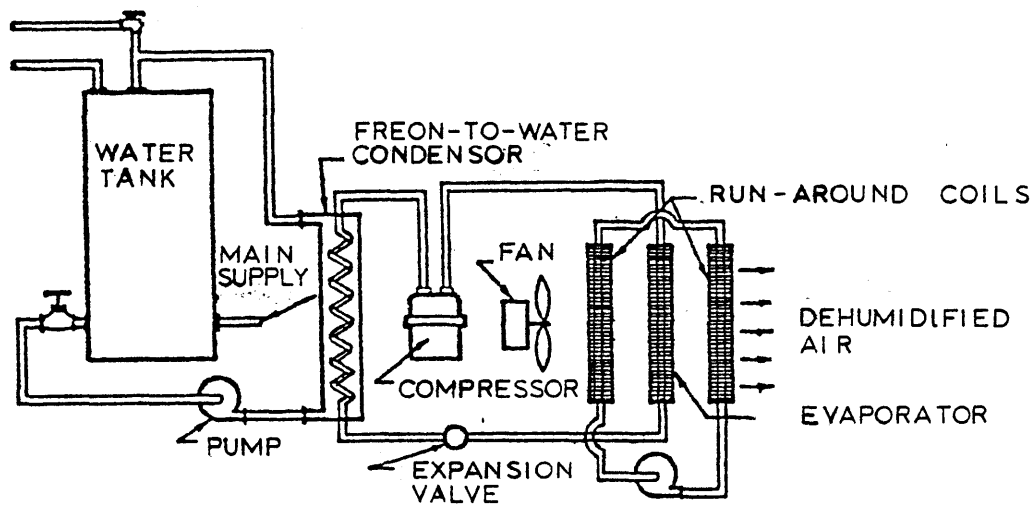


Fig. 5.14 Heat Pump DHW Heater with A Run-Around Coil

During the seasons when cooling is not needed, the heat pump DHW heater is operated with a water to water heat exchanger and the heat extracted from the cooling reservoir is utilized to meet the daily DHW needs. The average COP is approximately 3.5. In summer, however, the heat pump DHW heater is operated with an air to water run-around coil heat exchanger. The latent heat extracted from the air is used to heat the DHW. As for the sensible cooling load in summer, it is carried by the cooling capacity stored in the cooling reservoir.

This mechanical system does not include two different heat pumps. It consists of only one heat pump with an automatically or manually controled humidistat which determines whether an air to water or water to water heat exchanger should be in operation. Generally speaking, for Taipei the water to water heat exchanger is operated from Oct. to May, and the air to water heat exchanger is operated from June to Sept..

In June and Sept. when the latent heat load is not high enough to use the heat pump for dehumidification, the automatically controlled humidistat would switch the system operation from an air to water heat exchanger mode to a water to water heat exchanger mode. This system would then extract heat from the cooling reservoir rather than from the air in order to meet the daily DHW needs.

4.4.2 Annual Reservoir Temperature Profile

An MIT project using phase change material to store "cool" in a radiant ceiling showed the average temperature difference between the room air and the radiant panel ceiling surface is approximately 8°F . That means, for a typical room condition with 78°F and 50% RH, the chilled water, pumped from the cooling reservoir and circulated through the radiant panel ceiling, should not exceed 70°F ($78^{\circ}\text{F}-8^{\circ}\text{F}$) in order to insure its cooling capability.

If the water temperature in the reservoir goes below 40°F , the COP of the heat pump DHW heater will decrease significantly. This situation should be avoided for the sake of energy conservation.

From the above analysis, it can be concluded that the water temperature in the cooling reservoir should be in the range between 40°F and 70°F . Fig. 5.15 shows the yearly reservoir temperature profile for the typical design in this thesis. This profile is derived from the heat exchange calculations in Appendix B, and is determined by the DHW needs and sensible cooling discharge. Note that the 2.75% more cooling capacity left in the reservoir at the end of september is to account for the inaccuracy of hand calculations and to assure this system's capability to meet the summer cooling load.

4.4.3 Mechanical Loops

Figs. 5.16 and 5.17 respectively show the mechanical loops which are involved in the mechanical systems in summer

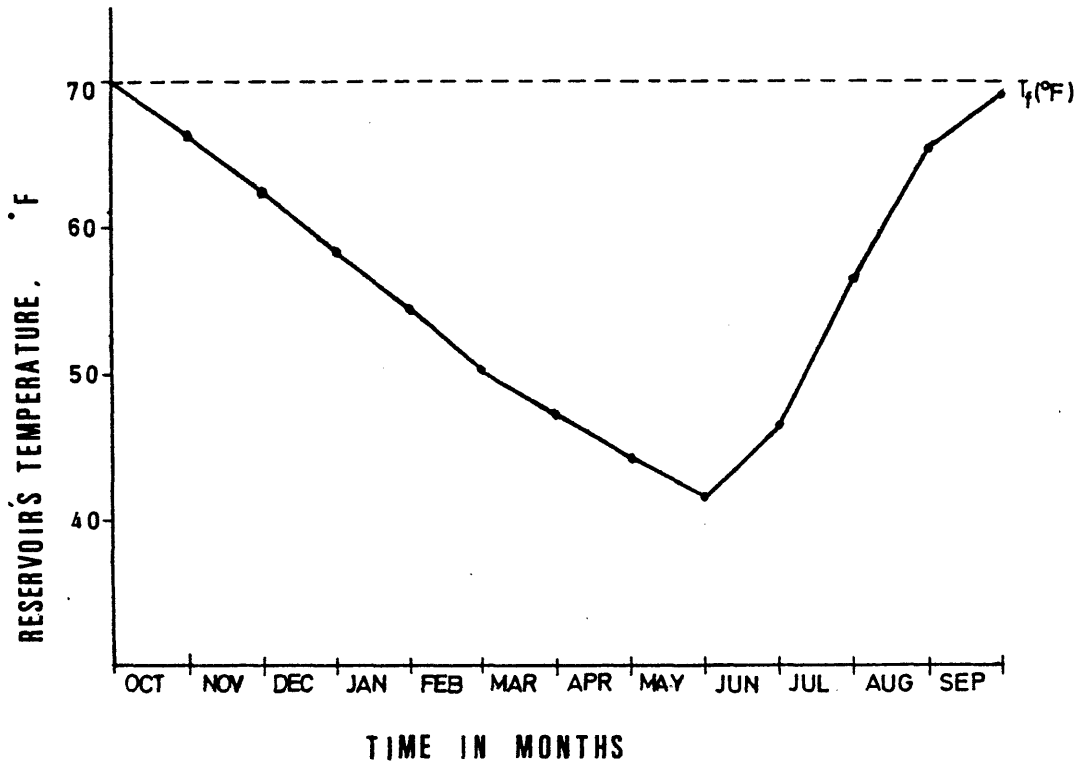


Fig. 5.15 Annual Cooling Reservoir Temperature Profile

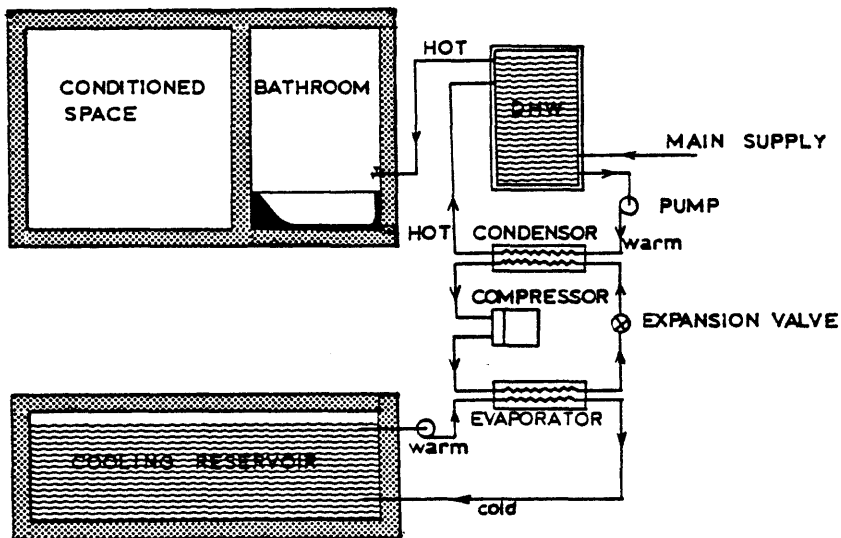


Fig. 5.16 Mechanical Loops for Non-Cooling Season Operation

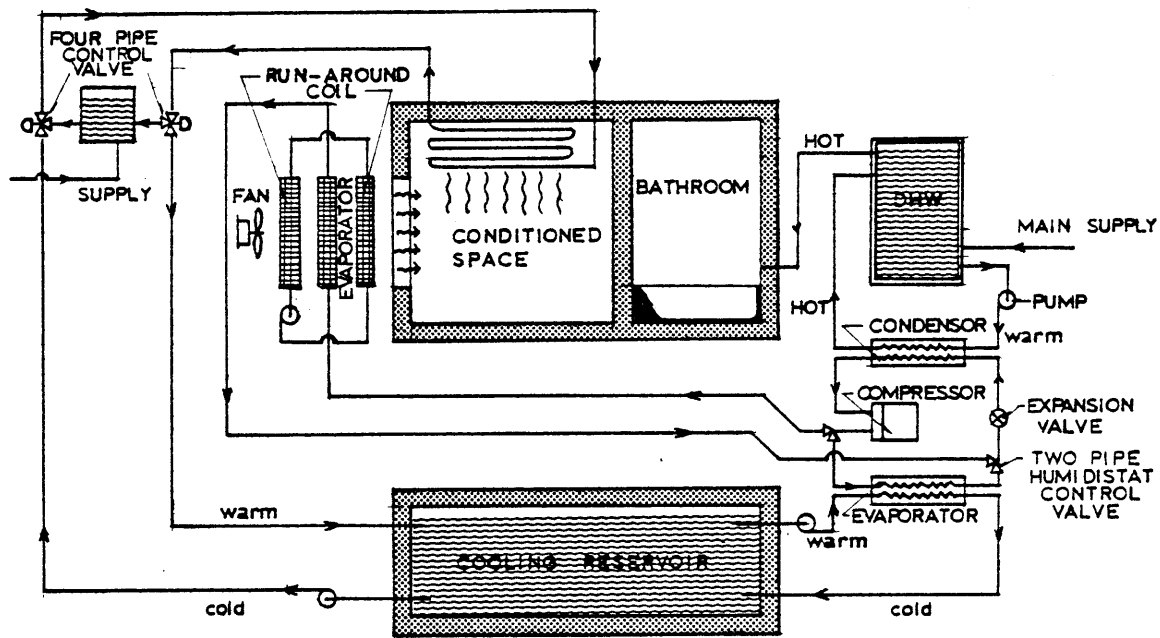


Fig. 5.17 Mechanical Loops for Cooling Season Operation

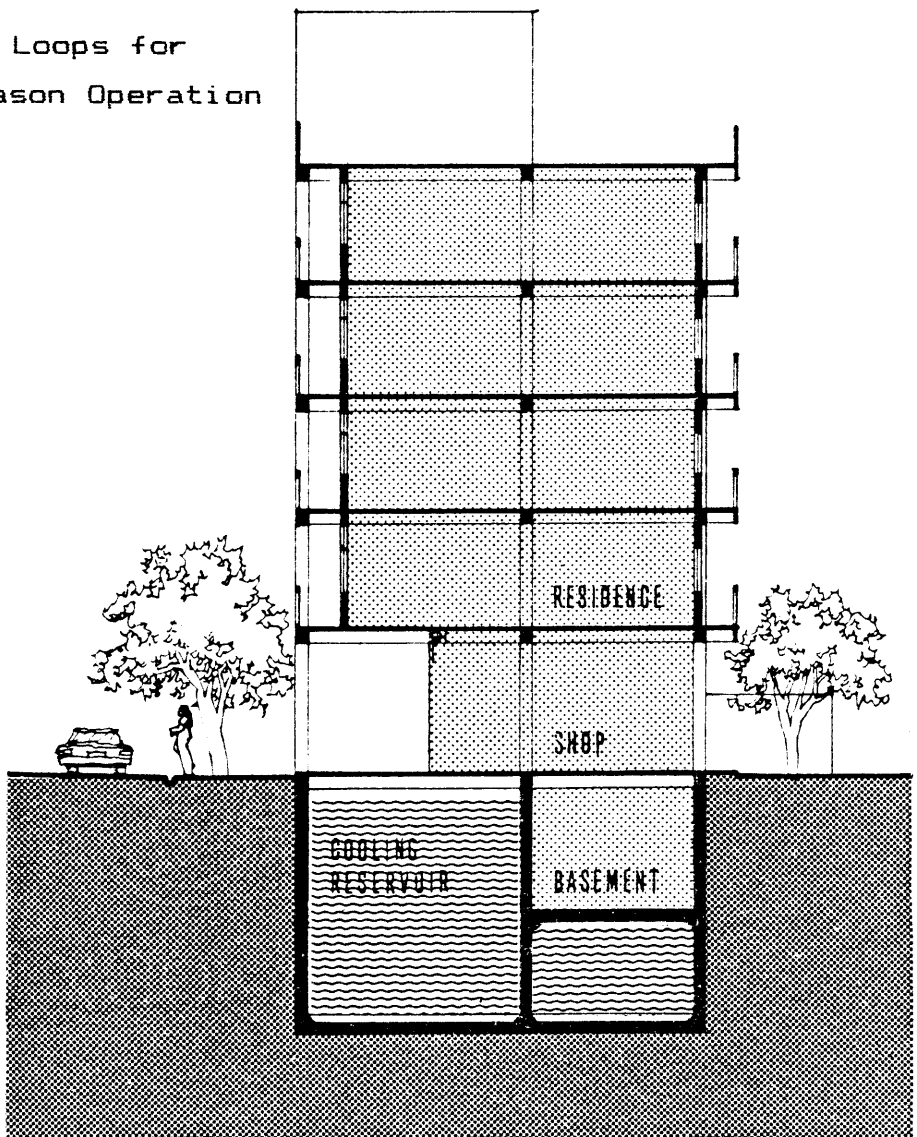


Fig. 5.18
The Perspective of Cooling Reservoir Application

and in non-cooling seasons.

5.0 ECONOMICAL ANALYSIS

5.1 Initial Cost

There are 3 major costs involved in the proposed cooling technique: the cost of cooling reservoir, the cost of heat pump DHW heater, and the cost of miscellaneous mechanical equipment such as piping, pumps, motor, heat exchangers, controls and sensors, etc..

The cost of miscellaneous mechanical equipment is small because no sophisticated devices are involved. The heat pump DHW heater can be purchased at a very low price (\$550-650 U.S. dollars), and is much cheaper than a solar DHW system (\$2500-3500 U.S. dollars). As far as the side benefit of its cooling capacity is concerned, the cost of a heat pump DHW heater is much less than that of a conventional air-conditioning system plus any DHW heater (solar heater, electrical heater, or gas propane heater).

Among the 3 major costs, the dominant one is the cost of a cooling reservoir. Its relatively high cost is due to its size and the large amount of insulation needed. However, because the reservoir sized for two families presented in this thesis is used as the floating foundation shown in the building construction as depicted in Fig. 5.3, the initial construction cost of it is almost the same as the cost of other types of foundations. There is, then, only a small increase in overall construction costs. From

the analysis above, a conclusion can be drawn that the extra initial cost of this cooling reservoir is fairly low assuming low cost insulation materials can be used for the reservoir.

5.2 Energy Savings

5.2.1 Domestic Hot Water Saving

The most popular DHW heater in Taipei is the gas-propane type. If it is replaced by a heat pump DHW heater with an average COP of 3.5 in non-cooling seasons and 5.0 in summer (with run-around coil), the annual cost saving for the typical 6-person extended family's demand for DHW is \$148 U.S. dollars, approximately 58% less than the original system. This result is derived in Appendix D.

5.2.2 Summer Air-Conditioning Saving

An ability to provide sensible as well as latent cooling is the side benefit of using the heat pump DHW heater. The sensible and latent cooling capacity can be regarded as a "free gift" from applying this system. Large energy or cost saving in cooling can be obtained because the conventional air-conditioning system is not in use. The total cost of using air-conditioning system with a typical COP of 2.0 for this typical family is \$127 U.S. dollars in Taipei as calculated in Appendix D. That means, this total operating cost is almost saved in one year because only the water pump energy is used for cooling when the low-cost

cooling technique is applied.

5.2.3 Total Yearly Saving

After we add the savings from the DHW heater and air-conditioning systems, the total saving in energy cost of a 6-person extended family is \$275 U.S. dollars (\$9890 N.T. dollars) for each year.

If we know precisely the total initial investment, (which depends on many different parameters, and as such is beyond the scope of this thesis) and present discount rate, the pay-back period can be derived.

5.3 Perspective

The cooling reservoir presented in this thesis is only sized for two family's cooling loads (Fig. 5.3). As it is used structurally as a floating foundation, it can be regarded as having very little initial cost as explained before. If the whole apartment (not including the first floor shopping area) needs to be cooled by this cooling technique, an additional reservoir of the same size must be added below the first one (Fig. 5.18). The cost of the second one however, is much greater than the first one from the construction point of view because it does not function as a structural component.

REFERENCES FOR CHAPTER V

1. J.M., Akridge, "Investigation of Passive Cooling Techniques for Hot-Humid Climates", (College of Architecture, Georgia Institute of Technology), chapter 8.
2. Ibid.
3. Obrecht, Malvern et. al., "Radiant Panel Ceilings : Painstaking Design Brings Maximum Comfort", Heat, Piping and Air-Conditioning, Vol 45, no.10, September 1973.
4. Ibid.
5. J.M., Akridge, "Investigation of Passive Cooling Techniques for Hot-Humid Climates", (College of Architecture, Georgia Institute of Technology), chapter 7.
6. Ibid.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

1.0 CONCLUSIONS

Passive cooling in hot, humid climates has proven to be the most difficult issue in environmentally positive building design. Most of the available passive cooling techniques discussed in Chapter II, section 3.0 -7.0, are not applicable to these climates. Convective cooling is the only technique which can provide comfort in spring and fall. If the natural ventilation through the building is not effective due to the prevailing wind pressure pattern; the wind-inducing devices presented in Chapter IV, section 3.0 are necessarily applied.

In summer, however, convective cooling is inappropriate during the daytime due to extremely high air temperatures and relative humidities. The building must then be cooled by mechanical systems such as conventional air-conditioning systems or preferably the low-cost cooling system presented in Chapter V.

Summer cooling loads can first be greatly reduced by a good building envelope design. The load control strategies mentioned in Chapter II such as a south-facing orientation, optimal shading design, external louvered blinds, reflective, light-colored surfaces, appropriate amounts of insulation, and a well sealed building are all compulsory to

the maximum reduction of heat gains by convection, conduction, and radiation.

The local weather data and the general design tools have been presented in Chapter III. It is necessary that they be understood and used by architects in order to achieve a climatically responsive building design.

The proposed design layout in Chapter IV not only provides the maximum cross-ventilation potential, and it also solves the problem of privacy interruption as it exists in the popular 5-story walk-up apartments of Taipei. Its similar outward appearance and construction makes it acceptable to both users and builders.

The heat pump DHW heater coupled with a seasonal cooling reservoir appears to have considerable potential as a cooling concept for hot, humid climates. It can provide both a summer cooling capability and satisfy the annual DHW needs. The yearly operating cost of this system is only about 40% that of a conventional air-conditioning system plus a propane DHW heater. The saving is approximately \$230 U.S. dollars per year for the 6-person extended family unit referred to in this thesis. This savings shows the significant value of applying this low-cost cooling technique.

The concept of using a radiant panel ceiling to provide sensible cooling is also better than any other space conditioning system as is indicated in the ASHRAE Handbook of Fundamentals. This superiority is due to the fact that

the cooling load in the building is dealt with directly and locally by the large area of radiant heat sink.

The economy of the proposed system, however, is also greatly dependent upon the initial cost of the cooling reservoir. The reservoir's volume for each family is about 40% of the family unit volume. This large volume would indicate a fairly high initial cost if the reservoir is not used as a floating foundation or if the insulation material is very expensive. More information about the real cost of a reservoir is necessary before a final evaluation of the economical feasibility of this system can be made.

2.0 RECOMMENDATIONS

It is recommended that the human comfort analysis presented by Givoni's psychrometric chart and Aren's bioclimatic chart be re-evaluated. The current analysis is based on the average experiences of Americans who live in temperate climates. This analysis has a dubious application to people living in other climate types. The re-evaluation of this analysis should be based directly on the experiences of people living in hot, humid climates.

It is recommended that a computer simulation be established for a more uniform and precise method of system performance analysis. In addition, the construction of a full scale model is encouraged to evaluate the computer predicted performance, to determine implementation problems, and to establish the limits of feasibility.

It is recommended that heat pumps run by gas or other fossil fuels be extensively developed to replace contemporary types of electrically run heat pumps. This strategy promises a great, even if temporary, reduction of operating costs because of the lower price of gas and other fossil fuels. In Taipei, the price of gas is 60% that of electricity, so that given the same amount of energy consumption, operating costs should be similarly reduced.

It is recommended that the mechanical equipment involved in the proposed low-cost cooling systems such as thermostats, humidistats, 4-pipe control valves, etc. be developed and produced locally rather than imported from foreign countries in order to reduce the overall initial cost. It is also recommended, furthermore, that the run-around coil on the heat pump DHW heater be further developed and evaluated towards optimizing the latent cooling potential and increasing the COP of the heat pump.

It is further recommended that wind tunnel tests be made to establish the potential convective cooling techniques, and to aid in the design of wind-inducing devices. These tests should be concerned with qualitative and quantitative issues. The qualitative issues can be tested by examining white smoke blown by the prevailing winds to illustrate the wind pressure patterns and to mark the route of air flow. The model is best made in dark colors to render a sharp color contrast with the white smoke. The quantitative tests should measure the relative

air speeds at different positions inside buildings versus the concomitant outdoor air speeds at same height. The positions tested should be on the human level and at the points where ventilation is needed.

APPENDIX A

COOLING LOAD CALCULATIONS

1.0 SOL-AIR TEMPERATURE

2.0 U-VALUES AND AREAS OF EXTERNAL WALLS, DOORS, AND WINDOWS

3.0 COOLING LOAD IN JUNE

3.1 Sensible Load

3.2 Latent Load

4.0 COOLING LOAD IN COOLING SEASONS

APPENDIX A

COOLING LOAD CALCULATIONS

1.0 SOL-AIR TEMPERATURE

The cooling load resulting from conduction heat gains through the external walls and roof relate to the concept of "sol-air temperature". The sol-air temperature is the outdoor air temperature corrected to account for the solar energy absorbed in the external surfaces. The ASHRAE Fundamentals develops a method to calculate the sol-air temperature as shown by equation A-1

$$T_{\text{sol-air}} = T_o + \alpha * I_t / h_o - \epsilon A R / h_o \quad (A-1)$$

Where: α = absorptance of the external surface for solar radiation.

I_t = total solar radiation incident on the external surface (Btu/hr-sqft).

h_o = coefficient of heat transfer by long wave radiation and convection at the external surface (Btu/hr-sqft-°F).

T_o = outdoor air temperature(°F).

ϵ = hemispherical emittance of the external surface.

ΔR = the difference between the long-wave radiation incident on the external surface from the sky and surroundings and the radiation emitted by a black body at outdoor air temperature (Btu/hr-sqft).

The sol-air temperatures in Table A.1 have been calculated under the following assumptions:

* $\Delta R/h_o = 0$ for vertical surface.

= 0.15 for light-colored surface.

$h_o = 3.5$ for average 6mph wind speeds.

T_o = obtained from Taipei's M.A.H.W.D.

I_t = The half-day heat gains (6am-12am) through a 0.125 inch sheet glass on the west-facing facade, obtained from 1977 ASHRAE Fundamentals, CHAPTER 26, table 20, are doubled to calculate the whole-day sol-air temperatures. These figures only account for the diffuse radiation gains because the direct beam radiation is totally shaded in this design.

2.0 U-VALUE AND AREAS OF EXTERNAL WALLS, DOORS, AND WINDOWS

a) $U_{wall} = 0.089$ (as calculated below)

External Wall Construction	R	U = 1/R
1) Outside surface (summer)	0.25	
2) Stucco, 1.5 cm	0.10	
3) Plywood, 1.5 cm	0.62	
4) Vapor barrier	-	
5) Mineral fiber, 5.5 cm	8.0	
6) Air space, 4 cm	1.01	
7) Common brick, 11 cm	0.44	
8) Plaster, 1.5 cm	0.10	
9) Inside surface (still air)	0.68	
(See Fig. 4.5)	11.20	0.089 Btu/ sqft-°F

$$A_{\text{wall}} = 474.54 \text{ ft}$$

$$b) U_{\text{window}} = 0.56 \text{ (double pane)}$$

$$A_{\text{window}} = 86.62 \text{ ft} + 71.0 \text{ ft} = 157.62 \text{ ft}$$

Front Facade Rear Facade

$$c) U_{\text{door}} = 0.01 \text{ (insulated R-10 door)}$$

$$A_{\text{door}} = 38.74 \text{ ft} + 38.74 \text{ ft} = 77.48 \text{ ft}$$

Front Facade Rear Facade

Since the operation schedules of the apartments are all the same, the heat exchange between the adjacent family

units is negligible and is not involved in the cooling load calculations. Such partition walls are therefore considered to be adiabatic.

3.0 COOLING LOAD IN JUNE

3.1 Sensible Load

$$\begin{aligned}\text{CLTD} &= T_{\text{average solair}} - T_{\text{interior}} \\ &= 85.5 - 78 = 7.5^{\circ}\text{F}\end{aligned}$$

1) R-11 External Wall

$$\begin{aligned}Q/\text{hr} &= U * A * \text{CLTD} \\ &= 0.089 * 474.54 * 7.5 \\ &= 316.8 \text{ Btuh}\end{aligned}$$

$$\begin{aligned}Q/\text{day} &= 316.8 \text{ Btuh} * 10 \text{ operation hrs/day} \\ &= 3,168 \text{ Btu/day}\end{aligned}$$

2) R-10 Door

$$\begin{aligned}Q/\text{hr} &= U * A * \text{CLTD} \\ &= 0.1 * 77.48 * 7.5 \\ &= 581 \text{ Btuh}\end{aligned}$$

$$\begin{aligned}Q/\text{day} &= 58.1 \text{ Btu/hr} * 10 \text{ operation hrs/day} \\ &= 581 \text{ Btu/day}\end{aligned}$$

3) Glass

a) Conduction heat gain:

$$\begin{aligned}Q/\text{hr} &= U * A * \text{CLTD} \\ &= 662 \text{ Btuh}\end{aligned}$$

$$\begin{aligned}Q/\text{day} &= 662 \text{ Btuh} * 10 \text{ operation hrs/day} \\ &= 6,620 \text{ Btu/day}\end{aligned}$$

b) Radiation heat gains:

with external louver

$$\begin{aligned} Q/\text{day} &= A * SC * SHGF \\ &= (157.62 - 21.52) \text{ ft} * (0.85 * 0.88 \\ &\quad * 0.15) * 179 \text{ Btu/ft half-day} * 2 \\ &= 5,469 \text{ Btu/day} \end{aligned}$$

without external louver

$$\begin{aligned} Q/\text{day} &= A * SC * SHGF \\ &= 21.52 \text{ ft} * (0.89 * 0.88) \\ &\quad * 179 \text{ Btu/ft half-day} * 2 \\ &= 5,762 \text{ Btu/day} \end{aligned}$$

4) Infiltration

$$\begin{aligned} Q/\text{hr} &= 1.10 * CFM * T \\ &= 1.10 * \frac{(8,630 * 0.5)}{60} * (85.5 - 78) \\ &= 593.3 \text{ Btuh} \\ Q/\text{day} &= 593.3 \text{ Btuh} * 10 \text{ operation hrs/day} \\ &= 5,933 \text{ Btu/day} \end{aligned}$$

5) Occupants

$$\begin{aligned} Q/\text{day} &= 225 \text{ Btu/person} * 3 \text{ people} \\ &\quad * 10 \text{ operation hrs/day} \\ &= 6,750 \text{ Btu/day} \end{aligned}$$

6) Kitchen

$$\begin{aligned} Q/\text{day} &= 18,600 \text{ Btu/day} * \frac{10 \text{ operation hrs}}{24 \text{ hrs/day}} \\ &= 7,750 \text{ Btu/day} \end{aligned}$$

7) Refrigerator

$$\begin{aligned} Q/\text{day} &= 3.4 * W \\ &= 3.4 * (1.4 \text{ kw/day} * 1,000 \text{ w/kw}) \\ &\quad * 10 \text{ operation hrs} \end{aligned}$$

$$24 \text{ hrs/day}$$

$$= 7,083 \text{ Btu/day}$$

8) T.V.

$$Q/\text{day} = 3.4 * W$$

$$= 3.4 \text{ Btu/w} * (1.4 \text{ kw/day} * 1,000 \text{ w/kw})$$

$$* \frac{2 \text{ hrs}}{6 \text{ hrs/day}}$$

$$= 1,586 \text{ Btu/day}$$

9) Laundry

$$Q/\text{day} = 3.4 * W$$

$$= 3.4 \text{ Btu/w} * (0.3 \text{ kw/day} * 1,000 \text{ w/kw})$$

$$= 1,020 \text{ Btu/day}$$

10) Lights

$$Q/\text{day} = 3.4 * w$$

$$= 3.4 \text{ Btu/w} * (1.2 \text{ kw/day} * 1,000 \text{ w/kw})$$

$$= 4,080 \text{ Btu/day}$$

11) Heat loss from DHW (R-11 insulation)

$$Q/\text{day} = U * A * T$$

$$= 0.09 * 36.4 * \frac{(79 + 60 + 79)}{2} - 78$$

$$= 1,025 \text{ Btu/day}$$

$$\text{Total} = 56,827 \text{ Btu/day}$$

$$\text{Monthly Total} = 56,827 * 30$$

$$= 1,704,810 \text{ Btu/month}$$

3.2 Latent Load

1) Infiltration

$$H/\text{hr} = 4,840 * \text{CFM} * W$$

$$= 4,840 \text{ Btu/min} * \frac{(8,630 / 2)}{60} * \frac{(129 - 72)}{7,000}$$

$$= 2,834 \text{ Btu/hr}$$

2) People

$$H/\text{day} = 229 \text{ Btu/h/person} * 3 \text{ people} * 10 \text{ hrs/day}$$

$$= 6,750 \text{ Btu/day}$$

$$\text{Total} = 35,090 \text{ Btu/day}$$

4.0 COOLING LOAD IN COOLING SEASONS

Using the same heat gain equations, the cooling load in July, August, and September, can be calculated. The results are shown in Table A.2.

Month	June			July			August			September		
Hour	Tair (F)	It Btu/ (ft- F)	Tsol- air (F)	Tair (F)	It Btu/ (ft- F)	Tsol- air (F)	Tair (F)	It Btu/ (ft- F)	Tsol- air (F)	Tair (F)	It Btu/ (ft- F)	Tsol- air (F)
12- 1am	76.9	0	77.6	79.9	0	79.9	79.9	0	79.9	77.4	0	77.4
1- 2am	76.5	0	76.5	79.3	0	79.3	79.4	0	79.4	77.0	0	77.0
2- 3am	76.1	0	76.1	78.8	0	78.8	79.0	0	79.0	76.4	0	76.4
3- 4am	75.8	0	75.8	78.3	0	78.3	78.6	0	78.6	76.2	0	76.2
4- 5am	75.6	0	75.6	77.9	0	77.9	78.3	0	78.3	75.7	0	75.7
5- 6am	75.6	0	75.6	78.0	0	78.0	78.2	0	78.2	75.7	0	75.7
6- 7am	76.6	7	76.9	79.0	6	79.3	79.1	2	79.2	76.5	0	76.5
7- 8am	78.8	18	79.6	81.7	18	82.5	81.6	15	82.2	77.3	10	77.7
8- 9am	82.0	26	83.1	85.3	26	86.4	85.0	24	86.0	82.2	19	83.0
9-10am	84.4	32	85.8	88.0	32	89.4	87.4	30	88.7	84.6	26	85.7
10-11am	85.8	36	87.3	89.7	36	91.2	88.9	35	90.4	86.0	31	87.3
11-12am	86.3	39	88.0	90.5	39	92.2	89.5	37	91.1	86.7	34	88.2
12- 1pm	86.5	43	88.3	90.6	43	92.4	89.7	41	91.5	86.8	37	88.4
1- 2pm	85.9	39	87.6	90.3	39	92.0	89.3	37	90.9	86.4	34	87.9
2- 3pm	84.9	36	86.4	89.4	36	90.9	88.4	35	89.9	85.5	31	86.8
3- 4pm	83.6	32	85.0	88.3	32	89.7	87.4	30	88.7	84.3	26	85.4
4- 5pm	82.6	26	83.7	87.2	26	88.3	86.3	24	87.3	82.7	19	83.5
5- 6pm	81.5	18	82.3	86.4	18	87.2	84.7	15	85.3	81.3	10	81.7
6- 7pm	80.4	7	80.7	84.1	6	84.4	84.4	2	84.5	80.2	0	80.2
7- 8pm	79.7	0	79.7	83.2	0	83.2	82.6	0	82.6	79.6	0	79.6
8- 9pm	79.1	0	79.1	82.6	0	82.6	81.9	0	81.9	79.0	0	79.0
9-10pm	70.6	0	78.6	81.9	0	81.9	81.4	0	81.4	78.5	0	78.5
10-11pm	78.1	0	78.1	81.2	0	81.2	80.9	0	80.9	78.1	0	78.1
11-12pm	77.6	0	77.6	80.5	0	80.5	80.3	0	80.3	77.7	0	77.7
	10 Hour Average		85.8 F	16 Hour Average		87.2 F	16 Hour Average		86.5 F	10 Hour Average		85.8 F

TABLE A-1

Load Source			Hourly Cooling Load Equation	Daily Cooling Load Btu/day			
				June	July	August	Sept.
External Gains	Wall		$Q = UA(CLTD)$	3168	6081	5608	3126
	Door		$Q = UA(CLTD)$	581	1116	1029	589
	Glass	Conduction	$Q = UA(CLTD)$	6620	12,710	11,721	6708
		W/external louver	$Q = A(S.C.)SHGF$	5469	5406	5008	4247
		Without external louver	$Q = A(S.C.)SHGF$	5762	5698	5280	4474
Infil-tration	Sensible		$Q = 1.10(CFM)\Delta T$	5933	11,185	10,315	6012
	Latent		$Q = 4840(CFM) W$	28,340	47,736	47,736	24,365
Internal Gains	Occupants	Sensible	$Q = 225 * \text{persons}$	6750	13,050	13,050	6750
		Latent	$Q = 225 * \text{persons}$	6750	13,050	13,050	6750
	Kitchen		-	7750	12,400	12,400	7750
	Refrigerator		$Q = 3.4W$	7083	11,333	11,333	7083
	T.V.		$Q = 3.4W$	1586	4760	4760	1586
	Laundry		$Q = 3.4W$	1020	1020	1020	1020
	Lights		$Q = 3.4W$	4080	10,200	10,200	4080
	Heat loss from DHW Tank		$Q = UA\Delta T$	1025	1906	1906	1025
Daily Total (one family)		Sensible		56,827	96,865	93,630	54,450
		Latent		35,090	60,786	60,786	31,115
Monthly Total (one family)		Sensible		1,704,810	3,002,815	2,902,530	1,633,500
		Latent		1,052,700	1,884,366	1,844,366	933,450
Summer Total (one family)		Sensible		9,243,655			
		Latent		5,754,882			
Monthly Total (two family)		Sensible		3,409,620	6,005,630	5,805,060	3,627,000
		Latent		2,105,400	3,768,732	3,768,732	1,866,900
Summer Total (two family)		Sensible		18,487,310			
		Latent		11,509,764			

TABLE A-2

APPENDIX B

HEAT EXCHANGE AND TEMPERATURE PROFILE OF THE COOLING RESERVOIR

1.0 GENERAL CONCEPTS

2.0 HEAT EXCHANGE

2.1 The Heat Extracted from the Reservoir to Meet DHW Needs

2.2 The Heat Absorbed from the Cooling Load of the Room

2.3 The Heat Transferred into the Reservoir from Surroundings

3.0 TEMPERATURE DIFFERENCE RESULTING FROM HEAT EXCHANGES

4.0 HEAT EXCHANGES AND TEMPERATURE PROFILE FOR A 2-FAMILY SIZED COOLING RESERVOIR

APPENDIX B

1.0 THE GENERAL CONCEPTS

The main energy exchanges involved in the cooling reservoir are described below.

- 1) the energy extracted from the reservoir- (Q_e) - to heat the DHW (negative value).- (Q_e)
- 2) the energy transferred from the surroundings (Q_s) through the walls and ceilings of the reservoir (positive value). - (Q_s)
- 3) the energy absorbed from the room during the discharge periods (positive value). - (Q_a)

The net monthly energy input or output depends upon the total value of Q_e , Q_s and Q_a , i.e. $Q_{net} = Q_e + Q_s + Q_a$. If the value of Q_{net} is positive, the temperature of the tank would increase due to the energy input. If Q_{net} is negative, then the temperature of the tank would decrease.

2.0 HEAT EXCHANGE

2.1 The Heat Extracted from the Reservoir to Meet DHW Needs

2.1.1 Non-Cooling Seasons

In non-cooling seasons, the DHW needs are supplied by the extraction of energy from the cooling reservoir through a water source heat pump. The energy extracted for each month can be estimated by equation B-1.

$$Q_e = \text{Trise} * C * \rho * V_t * \frac{100}{98} * \left(\frac{\text{COP}_w}{\text{COP}_w + 1} \right) * D \quad (\text{B-1})$$

Where: Trise = Temperature rise in DHW tank in a day

C = Specific heat of water 1 Btu/lb F

ρ = Water density 62.44 lb/cuft

Vt = Volume of the tank

$\frac{100}{98}$ = 2% heat loss to surrounding

COPw = The coefficient of performance of a water source heat pump

$\frac{\text{COP}_w}{\text{COP}_w + 1}$ = Ratio of heat exchange between cool source and hot source

since

$$\text{COP}_w = \frac{\text{Energy Extracted}}{\text{Energy Input}}$$

Energy Output = Energy Extracted

+ Energy Input

= Energy Input * (COPw + 1)

then

$$\frac{\text{Energy Extracted from Cool Source}}{\text{Energy Output to Hot Source}} = \frac{\text{COP}_w}{\text{COP}_w + 1}$$

For example, the energy extracted (Qe) from the reservoir for a 6-person extended family to heat DHW in a typical month is calculated using equation B-1.

Where: Vt = 120 U.S. gallons

= 16.042 cuft for 6-person needs

T = 60°F in general case

COPw= 3.5 for common water source heat pump

D = 30 days in a month

Qe = 60 °F/day * 1 Btu/lb- F * 62.44 lb/cuft

* 16.042 cuft * $\frac{100}{98}$ * $\frac{3.5}{3.5 + 1}$ * 30 days/month

= 1,430,946 Btu/month

For two families, Qe = 1,430,946 * 2

= 2,861,893 Btu/month

2.1.2 Cooling Seasons

In summer time , the latent heat in the air can be transferred into DHW by a heat pump with a run-around coil. If DHW needs can not be entirely supplied with this system due to a lack of latent load, then the remaining portion of these needs must be supplied by the extraction of heat from the cooling reservoir. The energy (Qe) extracted monthly from the reservoir can be derived by equation B-2 for the summer season.

$$Q_e = (T_{rise} * C * \rho * V_t * \frac{100}{98}) - (H_l * \frac{COP_a + 1}{COP_a}) * \frac{(COP_w)}{COP_w + 1} * D \quad (B-2)$$

Where: Hl = Daily latent load in summer

COPa= The coefficient of performance of an
air-to-water heat pump

If the latent load were 35,090 Btu/day in June, for the same 6-person family referred to above, and a heat pump with a run-around coil were used (COPa = 5), then the energy extracted from the cooling reservoir (Qe) can be derived by using equation B-2

$$\begin{aligned}
 Q_e &= (60 * 1 * 62.44 * 16.042 * 100/98) \\
 &\quad - (35,090 * 6/5) * \frac{3.5}{3.5 + 1} * 30 \\
 &= 448,426 \text{ Btu/month}
 \end{aligned}$$

$$\begin{aligned}
 \text{For two families, } Q_e &= 448,426 * 2 \\
 &= 896,852
 \end{aligned}$$

2.2 Heat Absorbed from the Cooling Load of the Room

In the summer when chilled water is circulated through the ceiling coils of the conditioned space, the heat absorbed by this water is circulated back into the reservoir. Thus it becomes a heat input into the cooling reservoir. The total amount of this heat (Qa) for each month is equal to the monthly total sensible cooling load of the conditioned space. i.e.

$$Q_a = \text{Monthly Total Cooling Load} \quad (\text{B-3})$$

2.3 Heat Transferred into the Reservoir from the Surroundings

The total monthly energy transferred into the reservoir (Q_s) from the surroundings can be approximated by equation B-4

$$Q_s = U * A * (T_{air} - T_{wa}) * 24 * D \quad (B-4)$$

Where: U = U-value of the walls and ceiling of reservoir
 A = Surface area of walls and ceiling of reservoir
 T_{air} = Outdoor monthly average air temperature
 T_{wa} = The monthly average water temperature in the reservoir
 24 = Total hours in a day
 D = The number of days in that month

For example, in November where $T_{air} = 68.11^{\circ}\text{F}$, and if $T_{wa} = 64.11^{\circ}\text{F}$, $U = 0.02$, and $A = 2,035$ sqft, the energy transferred from surroundings (Q_{in}) can be calculated using equation B-3:

$$\begin{aligned} Q_{in} &= U * A * (T_{air} - T_{wa}) * 24 * D \\ &= 0.02 * 2,035 * (68.11 - 64.11) * 24 * 30 \\ &= 117,216 \text{ Btu/month} \end{aligned}$$

3.0 TEMPERATURE DIFFERENCE ($^{\circ}\text{T}$) RESULTING FROM ENERGY EXCHANGES

The temperature rise or fall in the water of the reservoir for each month can be determined by equation B-5:

$$T = \frac{Q}{C * \rho * V} \quad (B-5)$$

Where: Q = Monthly energy input (positive value) or output
(negative value) to the reservoir (Btu/month)

C = Specific heat of water (1 Btu/lb-F)

= Water density (62.44 lb/cuft)

V = Reservoir volume (cuft)

For example, if the net energy exchange (Q_{net}) in March is -2,491,076 Btu/month (negative value means energy is extracted out of the reservoir), and the reservoir's volume is 11,309.5 cuft, then the temperature difference (°T) can be calculated using equation B-4:

$$T = \frac{-2,491,076}{1 * 62.44 * 11,309.5} = -3.53 (^{\circ}F)$$

The negative value of T means that the water temperature has decreased.

4.0 ANNUAL HEAT EXCHANGES AND TEMPERATURE PROFILE OF A COOLING RESERVOIR FOR TWO FAMILIES

The procedure to approximate monthly heat exchanges and temperature difference of a cooling reservoir is shown below:

- 1) Find the initial water temperature (T_i) in the cooling reservoir at the beginning of the month

which is equal to the final water temperature of last month.

- 2) Calculate the total monthly energy extracted (Q_e) from the cooling reservoir using equation B-1 in non-cooling seasons and equation B-2 in cooling seasons. (negative value)
- 3) Calculate the total monthly energy absorbed (Q_a) from the conditioned room, which is transferred into the reservoir, using equation B-3. (positive value)
- 4) Calculate the initial estimated net energy exchange for that month (Q_i) by adding Q_e and Q_a . i.e. $Q_i = Q_e + Q_a$
- 5) calculate the initial temperature difference of the water (T_i) for that month, resulting from the initial estimated net energy exchange using equation B-5. That is, substitute Q_{net} for Q .
- 6) Calculate the estimated monthly average water temperature (T_{wa}) by dividing T_i by 2 and adding this result to T_i . i.e. $T_{wa} = T_i + (T_i/2)$
- 7) Find the monthly average air temperature (T_{air}) from the weather data.
- 8) Calculate the total monthly energy transferred into the reservoir from surrounding (Q_s) using equation B-4.
- 9) Calculate the net monthly energy exchange (Q_{net}) of the reservoir by adding Q_i and Q_s . i.e. $Q_{net} = Q_i + Q_s$

10.) Calculate the net water temperature difference (T_{net}) for that month using equation B-5. Substitute Q_{net} for Q .

11.) Add T_i and T . The final water temperature at the end of the month (T_f) can then be obtained.

$$T_f = T_i + T_{net}.$$

Table B-1 gives the annual energy exchanges and temperature profile calculated by the above procedures. Note the charge cycle is from October to May and the discharge cycle is from June to September. The water temperature range in the reservoir is between 70 F and 40 F for the capability of cooling potential and for the efficiency of the mechanical system respectively.

Table B-1 is calculated for two 6-person family units. The reservoir's volume is 11309.9 cubic ft. The insulation amount applied to the reservoir is R-50 both in the walls and in the ceiling. The left cooling potential (Q_{left}) at the end of the discharge periods (Sept.) can be estimated by equation B-5 where:

$$T = Q / C . V$$

$$\text{i.e. } (70 - 69.20) = Q_{left} / (1 \times 62.44 \times 1309.5)$$

$$Q_{left} = 508439$$

Divide this remaining cooling potential (Q_{left}) by the total summer sensible cooling loads, and we can conclude how much more percentage of cooling capability which this

low-cost system can provide for.

This percentage is derived below:

$$\begin{aligned}
 \% &= Q_{\text{remain}} / \text{total summer sensible cooling load} \\
 &= 508439 / 3409620 + 6005630 + 5805060 + 3267000 \\
 &\quad \quad \quad (\text{June}) \quad \quad (\text{July}) \quad \quad (\text{August}) \quad (\text{Sept}) \\
 &= 508439 / 18487310 = 0.6275 \\
 &= 2.75\%
 \end{aligned}$$

Month/Step	1	2	3	4	5	6	7	-	8	9	10	11
	Ti	Qe	Qa	Qinet	Ti	Twa	Tair	Tair - Twa	Qs	Qnet	Tnet	Tf
	(F)	Btu/ (Month)	Btu/ (Month)	Btu/ (Month)	(F)	(F)	(F)	(F)	Btu/ (Month)	Btu/ (Month)	(F)	(F)
Oct	70.00	-2,957,400	0	-2,957,400	-4.188	67.91	75.3	7.41	224,078	-2,733,322	-3.87	66.13
Nov	66.13	-2,862,000	0	-2,862,000	-4.053	64.11	68.1	4.00	117,216	-2,744,784	-3.89	62.24
Dec	66.24	-2,957,400	0	-2,957,400	-4.188	60.15	62.8	2.65	80,244	-2,877,158	-4.07	58.17
Jan	58.17	-2,975,400	0	-2,975,400	-4.188	56.08	60.0	3.92	118,822	-2,838,578	-4.02	54.15
Feb	54.15	-2,671,200	0	-2,617,200	-3.783	52.26	60.5	8.24	225,367	-2,445,833	-3.46	50.69
Mar	50.69	-2,957,400	0	-2,957,400	-4.188	48.60	64.0	15.40	466,324	-2,491,076	-3.53	47.17
Apr	47.17	-2,862,000	0	-2,862,000	-4.053	45.14	71.4	26.26	769,523	-2,092,477	-2.96	44.21
May	44.21	-2,957,400	0	-2,957,400	-4.188	42.12	75.4	33.28	1,007,745	-1,949,655	-2.76	41.44
Jun	44.14	+ 885,720	+3,409,620	+2,522,100	+3.570	42.23	80.4	37.17	+1,089,230	+3,611,330	+5.11	46.55
Jul	46.55	0	+6,005,630	+6,005,630	+8.500	50.80	83.8	33.00	999,266	+7,004,896	+9.92	56.47
Aug	56.47	0	+5,805,060	+5,805,060	+8.220	60.58	83.4	22.82	691,007	+6,496,067	+9.20	65.67
Sep	65.67	-1,110,180	+3,267,000	+21,756,820	+3.050	67.20	80.5	13.30	389,743	+2,546,563	+3.61	69.28

TABLE B

APPENDIX C

GROUND TEMPERATURE AT DIFFERENT DEPTHS

The ground temperature at any depth and at any time of the year can be predicted by equation C-1

$$T(x,t) = T_m - A_s * e^{-x \left(\pi / 365 \alpha \right)^{1/2}} * \cos \frac{360}{365} \left(* \left(t - t_o - \frac{x}{2} \left(\frac{365}{\pi * \alpha} \right)^{1/2} \right) \right)$$

Where: $T(x,t)$ = Ground temperature at depth x (ft) and day t
(°F)

T_m = Annual mean ground temperature (°F)

A_s = Annual surface temperature amplitude ($x=0$) (°F)

α = Ground diffusivity (sqft/day)

t = Day of the year

t_o = Day of minimum surface temperature

The ground temperatures in Table C have been calculated on the basis of the following parameters:

$T_m = 73.1$ °F (From Central Weather Bureau R.O.C.)

$A_s = 12.77$ °F

$\alpha = 0.6$

APPENDIX D

ECONOMICAL ANALYSIS

1.0 DHW SAVING

1.1 Annual DHW Energy Needs

1.2 Annual Operating Cost of a Propane DHW Heater

1.3 Annual Operating Cost of a Heat Pump DHW Heater

1.3.1 Energy Consumption in Non-Cooling Seasons

1.3.2 Energy Consumption in Cooling Seasons

1.3.3 Annual Energy Consumption

1.3.4 Annual Operating Cost

1.4 Annual DHW Savings

2.0 COOLING SYSTEM SAVINGS

2.1 Operating Cost of an Air-conditioning System

(COP = 2)

2.2 Operating Cost of the Proposed Low-Cost Cooling system

2.3 Annual Cooling System Savings

3.0 TOTAL ANNUAL OPERATING COST SAVINGS

3.1 Annual Operating Cost of the Old System

3.2 Annual Operating Cost of the Newly Proposed System

3.3 Total Annual Savings in Amount and as a Percentage of the Former Operating Cost.

1.0 DHW SAVINGS

1.1 Annual DHW Energy Needs

For a typical 6-person family, the DHW needs are 120 U.S. gallons per day (20 gallons per person). The rise in water temperature is generally assumed to be 60°F. The total yearly DHW energy needs (Qn) can be estimated by equation D-1

$$Q_n = C * \rho * V * T * \frac{100}{98} * D \quad (D-1)$$

Where C = specific heat of water (1 Btu/lb-°F)
 ρ = Water density (62.44 lb/cuft)
V = Tank volume (120 gallons = 16.042 cuft)
T = Water temperature rise for each day (60°F/day)
 $\frac{100}{98}$ = Number of days when the systems is in operation (365 days/year)
D = 2% heat loss

$$\begin{aligned} \text{Therefore } Q_n &= 1 \text{ Btu/lb-}^\circ\text{F} * 62.44 \text{ lb/cuft} * 16.042 \text{ cuft} \\ &\quad * 60^\circ\text{F/day} * 100/98 * 365 \text{ days/year} \\ &= (61,320.27 \text{ Btu/day}) * 365 \text{ days/year} \\ &= 22,384,090 \text{ Btu/year} \end{aligned}$$

1.2 Annual Operating Cost of a Propane DHW Heater

For an 80% efficient propane DHW heater, one cubic foot of gas can provide 800 Btu of heat, that means, 1 cubic

meter can provide 28,252 Btu. The current gas price in Taipei is \$11.21 N.T. dollars per cubic meter. The annual cost (C), therefore of a propane DHW heater is derived as shown below:

$$\begin{aligned}
 C_{\text{gas}} &= \frac{Q}{28,252 \text{ Btu/m}^3} * \$11.21 \text{ Btu/m}^3 \\
 &= \frac{22,384,090}{28,252} * \$11.21 \\
 &= 8,882 \text{ N.T. dollars} \\
 &= 247 \text{ U.S. dollars}
 \end{aligned}$$

1.3 Annual Operating Cost of a Heat Pump DHW Heater

1.3.1 Non-Cooling Seasons (from October to May)

- a) Energy consumption by a water source Heat pump with an average COP 3.5.

$$Q = \frac{Q_{\text{DHW/8 months}}}{\text{COP}_w + 1}$$

Where: $Q_{\text{DHW/8 months}}$ = The DHW needs for 8 months

COP_w = Average COP of a water source heat pump
= 3.5

The Total DHW energy needs can be derived using equation D-2

$$\begin{aligned}
Q_{DHW}/8 \text{ months} &= C * \quad * V * \quad T * \frac{100 * 243 \text{ day/months}}{98} \\
&= 1 * 62.44 * 16.042 * 60 * \frac{100 * 243}{98} \\
&= 14,902,284 \text{ Btu/ 8 months}
\end{aligned}$$

Therefore

$$\begin{aligned}
Q &= 14,902,284 / (3.5 + 1) \\
&= 3,311,619 \text{ Btu/8 months.....(1)}
\end{aligned}$$

- b) Energy consumption by a water pump with 0.1 horsepower (hp) per hour can be calculated using equation D-1

$$Q_{wp} = 0.1 \text{ hp} * \text{Hop/day} * D * 745.7 \text{ w/hp} * 3.412 \text{ Btu/w} \quad (D-1)$$

Where: Hop : Water pump operating hours

D : Total days in operation

The water pump operating hours can be estimated by equation D-2

$$\text{Hop} = \frac{\text{DHW needs}}{\text{PHLD} * \left(\frac{\text{COP}_w + 1}{\text{COP}_w} \right)} \quad (D-2)$$

where: PHLD : Peak hourly latent load

COP_w : COP of a water source heat pump (3.5)

Peak hourly latent load can be approximated below:

$$\begin{aligned}
 \text{PHLD} &= \frac{\text{latent Load in July}}{16 \text{ Operating hrs/day} * 31 \text{ days/month}} \\
 &= \frac{1,884,366}{16 * 31} = 3,799.125 \\
 &= 4,000 \text{ Btu/hr (for a factor of savings)}
 \end{aligned}$$

Therefore, the daily operating hours can be calculated using equation D-2

$$\begin{aligned}
 \text{Hop/day} &= \frac{\text{DHW needs/day}}{\text{PHLD} * \left(\frac{\text{COP}_w + 1}{\text{COP}_w} \right)} = \frac{61,329 \text{ Btu/day}}{4,000 * \left(\frac{3.5 + 1}{3.5} \right)} \\
 &= 11.9 \text{ Hrs} = 12 \text{ Hrs}
 \end{aligned}$$

The energy consumption by a water pump in these periods would then be derived using equation D-1

$$\begin{aligned}
 Q_{wp} &= 0.1 \text{ hp} * 12 \text{ hrs/day} * 243 \text{ days/8 months} \\
 &\quad * 745.7 \text{ w/hp} * 3.412 \text{ Btu/w} \\
 &= 741,962 \text{ Btu} \dots \dots \dots (2)
 \end{aligned}$$

As a result, the total energy consumption in the sum of (1) and (2)

$$\begin{aligned}
 \text{i.e. } Q_{nc} &= (1) + (2) \\
 &= 3,311,619 + 741,926 \\
 &= 4,053,545 \text{ Btu} \dots \dots \dots (3)
 \end{aligned}$$

1.3.2 Energy Consumption In Cooling Systems (From June to Sept.)

1) June

- a) Energy consumption by a heat pump with run-around coil (COPa = 5)

$$Q = \frac{\text{monthly latent load}}{\text{COPa}} = \frac{1,052,700}{5}$$

$$= 210,540 \dots \dots \dots (4)$$

- b) Energy consumption by a water source heat pump (COPw = 3.5)

$$Q = \frac{\text{monthly DHW needs}}{\text{COPw} + 1}$$

$$- \frac{\text{monthly latent load} * \left(\frac{\text{COPa} + 1}{\text{COPa}} \right)}{\text{COPw} + 1}$$

$$= \frac{1,839,788 - 1,263,240}{4.5} = \frac{576,540}{4.5}$$

$$= 128,121 \dots \dots \dots (5)$$

- c) Energy consumption by a water pump (o.i hp/hour) can be estimated by equation D-3

$$Q_{wp} = 0.1 \text{ hp} * \text{Hop/month} * 745.7 \text{ w/hp} * 3.412 \text{ Btu} -$$

$$(D-3)$$

Where: Hop : Water pump operating hours

Hop is calculated using equation D-2

$$\text{Hop} = \frac{576,548}{4,000 * \left(\frac{3.5 + 1}{3.5} \right)} = 112 \text{ hrs}$$

Therefore

$$\begin{aligned} Q_{wp} &= 0.1 * 112 * 745.7 * 3.412 \\ &= 28,496 \text{ Btu/month} \dots\dots\dots (6) \end{aligned}$$

The total energy consumption in June is the sum of (4), (5), and (6)

$$\begin{aligned} \text{i.e. } Q &= (4) + (5) + (6) \\ &= 210,540 + 128,121 + 28,496 \\ &= 367,157 \text{ Btu} \dots\dots\dots (7) \end{aligned}$$

2) July

Energy consumption by a heat pump DHW heater with a run-around coil (COP = 5)

$$\begin{aligned} Q &= \frac{\text{monthly latent load}}{\text{COP}} = \frac{1,884,366}{5} \\ &= 376,873 \text{ Btu/month} \dots\dots\dots (8) \end{aligned}$$

3) August

With the same procedure as calculations in July, the total energy consumption is estimated to be 376,873 Btu.. (9)

4) September

With the same procedure as calculations in June, the total energy consumption is estimated to be 382,233 Btu.. (10)

The total energy consumption for a heat pump DHW heater in cooling season (Q_c) would then be the sum of (7), (8), (9), and (10). i.e.

$$Q_c = 367,157 + 376,873 + 376,873 + 382,233$$

$$= 1,512,136 \text{ Btu}$$

1.3.3 Annual Energy Consumption (Qt)

This consumption is the sum of Qnc and Qn

$$Q_t = Q_{nc} + Q_n$$

$$= 4,053,545 + 1,512,136 = 5,565,681 \text{ Btu}$$

1.3.4 Annual Operating Cost (Ct-heat pump)

$$C_{t\text{-heat pump}} = \frac{\text{Current electricity price/KW} * Q_t}{3,412 \text{ Btu/KW}}$$

$$= \frac{\$2.25 \text{ N.T. dollars/KW} * 5,565,681 \text{ Btu}}{3,412 \text{ Btu/KW}}$$

$$= \$3,670 \text{ N.t. dollars}$$

$$= \$102 \text{ U.S. dollars}$$

1.4 Annual DHW Savings

The annual savings by substituting a heat pump DHW heater for a gas-propane heater is calculated below:

$$C_{\text{saving}} = C_{\text{gas}} - C_{\text{heat-pump}} = 8,882 - 3,670$$

$$= 5,212 \text{ N.T. dollars}$$

$$= 148 \text{ U.s. dollars}$$

Total percentage of operating cost savings, for DHW is shown below:

$$\frac{\$52,12}{8,882} = 58.7\%$$

$$\begin{aligned}
\text{Cwp-cooling} &= \text{Price of electricity} * \text{Qwp-cooling} \\
&= \frac{\$2.25 \text{ N.T.dollars} * 405,057 \text{ Btu}}{3,412 \text{ Btu/KW}} \\
&= \$267 \text{ N.T. dollars}
\end{aligned}$$

2.3 Annual Cooling System Savings

This saving is derived by the difference between Cair-conditioning and Cwp-cooling. i.e.

$$\begin{aligned}
\text{Cs} &= \text{Cair-conditioning} - \text{Cwp-cooling} \\
&= 4,945 - 267 \\
&= \$4,678 \text{ N.T. dollars}
\end{aligned}$$

3.0 TOTAL ANNUAL OPERATING COST SAVINGS

3.1 Annual Operating Cost of the Conventional Systems (Co)

This cost is the sum of Cgas and Cair-conditioning

i.e. $\text{Co} = \text{Cgas} + \text{Cair-conditioning}$

$$\begin{aligned}
&= 8,882 + 4,945 \\
&= \$13,827 \text{ N.T. dollars}
\end{aligned}$$

3.2 Annual Operating Cost of the Proposed New Systems (Cn)

This cost is the sum of Cheat pump and Cup-cooling

i.e. $\text{Cn} = \text{Cheat pump} + \text{Cwp-cooling}$

$$\begin{aligned}
&= 3,670 + 267 \\
&= \$3,937 \text{ N.T. dollars}
\end{aligned}$$

3.3 Total Annual Savings in Amount (C) and in

Percentage (%)

1) The Amount

$$C = c_o - C_n = 13,827 - 3,937$$

$$= 9,890 \text{ N.T. Dollars}$$

$$= 275 \text{ U.S. dollars}$$

2) The Percentage of the Former Operational Cost

$$\frac{C}{C_o} * 100\% = \frac{9,890}{13,827} = 71.5\%$$

BIBLIOGRAPHY

A PASSIVE COOLING COOLING DESIGN FOR MULTIFAMILY RESIDENCES IN HOT, HUMID CLIMATES

1. Akridge, J.M., "Investigation of Passive Cooling Techniques for Hot-Humid Climates", College of Architecture, Georgia Institute of Technology, October, 1982.
2. Arens, E., Gonzalez, R., Berglund, L., McNall, P. E., Zeren, L., "A NEW BIOCLIMATIC CHART FOR PASSIVE SOLAR DESIGN", Proceedings of the 5th National Passive Solar Conference, Volume 5.2, (Amherst, Mass, American Sections of Ises, October 1980.)
3. Benton, C. C., and Johnson, T. E. "OFF-PEAK COOLING USING PHASE CHANGE MATERIAL", Department of Architecture, Massachusetts Institute of Technology, Cambridge, Mass., December 14 1979.
4. Bowen, Arthur, Clark, Eugene and Kennethlabs. Passive Cooling, International Passive and Hybrid Cooling Conference, Miami Beach, 1981.
5. Construction Codes, Taipei, Taiwan, R.O.C., 1977.
6. Coonley, Douglas R. Wind, The Franklin Institute Press, 1979.
7. Devol, Jamie. "NATURAL SPACE CONDITIONING FOR A TALL RESIDENTIAL STRUCTURE." Master Thesis, Massachusetts Institute of Technology School of Architecture, 1980.
8. Gilman, Stanley F., Solar Energy Heat Pump Systems for Heating and Cooling Buildings, The Pennsylvania State University, 1975.
9. Givoni, B. Man, Climate and Architecture, Applied Science Publishers, LTD., London 1976.
10. Greeley, Richard S., Duelllette, Robert P. and Cheremisinoff, Paul N.. Solar Heating and Colling of Building, Ann Arbor Science Publishers, INC., 1981.
11. Handa, Kamal. Wind Induced Natural Ventilation, Swedish Council for Building Research, Sweden, 1979.
12. Heap, R. D., Heat Pumps, E.& F. N. Spon LTD., 1979.
13. Himmelman, William A. Solar Engineering for Domestic

Buildings, Marcel Dekker, INC, 1980.

- 14 Johnson, Timothy E. Solar Architecture: The Direct Gain Approach, McGraw-Hill Book Company.
- 15 Kannengeg, Susan. "ESTIMATING SOLAR RADIATION FOR BUILDING DESIGNS." Solar Design Series.
- 16 Loftness, Vivian. "CLIMATE/ENERGY GRAPHICS.", Journal of Teaching Passive Design in Architecture, 1981.
- 17 Labs, Kenneth, "UNDERGROUND BUILDING CLIMATE. "Solar Age, 1979.
- 18 Lau, Andrew. "HOW TO DESIGN FIXED OVERHANGS." Solar Age, 1983.
- 19 McQuiston, Faye C. and Parker, Jerald D.. Heating, Ventilating, and Air Conditioning: Analysis and Design. John Wiley and Sons, INC, 1982.
- 20 Nazria, Edward. The Passive Solar Energy Book, Library of Congress, 1979.
- 21 Moore, Dennis. "HEAT PUMPS: HOT WATER AT THE RIGHT PRICE?" Solar Age, November 1981.
- 22 Neufert, Ernest. Architects' Data, Granada Publishing, 1980. Solar Heating and Cooling of Residential Buildings: Design of Systems, Solar Energy Applications Laboratory, Colorado State University, 1980.
- 23 Niles, Philip W. B. and Haggard, Kenneth L. Passive Solar Handbook 1980.
- 24 Obrecht, Malvern F. and Salinger, Robert. J. "RADIANT PANEL CEILINGS." Heating/Piping/Air Conditioning, September, 1973.
- 25 Olgyay, Victor. Design with Climate: Bioclimate Approach to Architectural Regionalism, Princeton University Press, 1963.
- 26 Packard, Robert T. Ramsey/Sleeper Architectural Graphic Standards, The American Institute of Architects, 1981.
- 27 Staples, Bay. Philadelphia Climate Data Digest, July, 1980.
- 28 Wittenberg, Gordon and Turner, William. "CLIMATE BASED PASSIVE COOLING", Journal of Teaching Passive Design in Architecture, 1981.